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THE INTERNAL COMBUSTION ENGINE

VOL. II

THE AERO-ENGINE

BY

D. R. PYE

DEPUTY DIRECTOR OF SCIENTIFIC RESEARCH, AIR MINISTRY; FORMERLY FELLOW AND LECTURER, TRINITY COLLEGE, CAMBRIDGE, AND FELLOW OF NEW COLLEGE, OXFORD

WITH A CHAPTER ON

THE AEROPLANE AND ITS POWER PLANT

BY

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PREFACE

ANY study of the special features of the aero-engine should enable the reader to see it against a background of the circumstances in which it is designed to work. Failing to find any book which made use of the modern aerodynamic theory to provide just the background required, I asked my friend Mr. W. S. Farren if he would fill the gap by contributing a special chapter on 'The aeroplane and its power plant'. This he has most kindly done, and there can be no doubt that, apart from its own intrinsic interest, Chapter II must greatly enhance the value of the rest of the book to the reader who is interested primarily in the engine. To Mr. Farren, therefore, first and foremost, my thanks are due. To Mr. Ricardo, also, and to Dr. A. A. Griffith, I am indebted for valuable criticism; and to the many others, too numerous to mention, from whom in almost daily discussion I have acquired much of the knowledge and such critical faculty as has gone to the writing of this book.

The substance of Chapters VI and VII has appeared previously in the form of a series of articles published in *Aircraft Engineering*.

D. R. P.

LONDON, 1934.

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I

INTRODUCTORY

Art. 1. *Aero-engines and others.*

The contents of the present volume will relate primarily to the aero-engine, and many of the problems to be discussed arise only from the special conditions in the air or from the exceptional speeds and cylinder conditions which are characteristic of the type. On the other hand, there is some truth in the saying that aeronautical engineering is ordinary engineering made more difficult, and especially in connexion with engines; so that many of the conclusions reached will be of quite general applicability to other forms of internal combustion engine.

Except in certain special features of its design the aero-engine of to-day is in all essentials the same as any other high-speed internal combustion engine which uses petrol as fuel. It is the aristocrat, or rather, perhaps, the athlete, of its species, with no superfluous tissue and every part fine-drawn to put forth its maximum effort and to endure for long periods under those severe conditions; but every individual part, even to auxiliaries like water- and oil-pumps, has a prototype in the automobile engine.

In discussing the aero-engine, we shall treat of power output per cylinder pressed to a point where the problems of heat-flow and inertia are of peculiar difficulty; but all conclusions as to the fundamental conditions which govern the power output of an engine will be equally applicable to engines where the working conditions are less arduous. Many of the problems of lubrication and overheating associated with the piston, for example, are common to most types of internal combustion engine; as are those, also, of the carburettor and of combustion in the cylinder.

It will be assumed that the reader is already familiar in a general way with the different types of aero-engines, 'in-line' and 'radial', water-cooled and air-cooled, and all purely descriptive matter is therefore omitted. Nothing more than a knowledge of the mere outlines of design will be called for, however, and where details become important in the discussion of principles these will be referred to and illustrated in their proper place.

An exception is made in regard to methods of cylinder construction. Some of these are described later in the present chapter, as illustrating the resources of the designer in the face of a demand for lightness, and because they are quite peculiar to aero-engines as a

class. Pistons, valves, and other features will come up for description in connexion with the problems of heat-flow and of cylinder cooling; and carburettors, superchargers, etc., in the chapters devoted to the principles of their operation.

Advances in the art of engine design and construction may usefully be considered under the four headings of

(*a*) Increase of power output per unit of weight.
(*b*) Improvement in fuel economy.
(*c*) Reduction of frontal area for a given power output.
(*d*) Increase of reliability and length of working life.

In the early days of flying the reduction of engine weight was of paramount importance, and it was the ability of the internal combustion engine to deliver power at a weight of about 6 lb. per B.H.P. which first made flying possible. Nothing but short flights at comparatively low speeds were contemplated, so that the problems of fuel economy, low frontal area, and length of working life were of minor importance.

Since those days the improvement of materials and of design has progressed to such a degree that in 1931 the winning Schneider Trophy Seaplane achieved a speed of 408·8 m.p.h., and was propelled by an engine which could deliver 2,350 B.H.P. for a total weight of 1,630 lb. Such a phenomenal power output is only possible, of course, with some sacrifice of reliability. The working life of the engine under those conditions would be no more than about four hours; but there are plenty of engines to-day which can show a weight of less than 2 lb. per B.H.P. at their full rated output; and capable under normal operating conditions of a life of 500 hours between overhauls.

The aim of lightness may now be said to have been so far achieved that the possible further gains in this direction are of minor importance, and aero-engine development is heading rather towards reducing frontal area, improving fuel economy, and towards an improved reliability both for the engine itself and for the no less important details of its installation in the aeroplane. To this effort towards reliability there is no end, short of perfection.

An advance in any one of the four directions mentioned above demands certain features now fairly clearly understood, but in the face of the past rapid progress further general advance must tend to be more difficult, and moreover the requirements for obtaining the best results in one direction are often mutually incompatible with the highest standard in the others. At the present day the problem of improved engine design, therefore, takes the form of striking a judicious and well-balanced compromise between conflicting

demands. And progress in one direction must needs be bought more and more at the price of sacrifice in another.

A good deal was written and argued in early days about the cylinder size to give minimum weight, but the results of discussions based upon first principles have, in this matter, little relation to practice. The discussions have to be based upon assumptions such as that the dimensions, and therefore weights, of the various parts can be reduced in proportion to an engine's overall size, and to the gas forces involved. This is by no means true, however, because so large a part of the necessary weight of a light engine rests upon requirements other than those of calculable strength—upon rigidity, for example, and heat conductivity, as well as upon all the structural features connected with auxiliaries like water-pumps and magnetos, which do not lend themselves readily to calculation.

It does not require a mathematical argument based on first principles to convince one that the lightest possible engine would be one with a large number of pistons connected to the minimum possible number of cranks, because the heaviest single features of an engine are the crankshaft and crankcase. As it happens, the radial engine affords a possibility of combining a short, light, crankshaft of one or two cranks only, with a very compact and rigid type of crankcase, carrying a large number of cylinders. In its air-cooled form the type also provides very uniform cooling for all the cylinders when exposed to an airstream in a direction parallel to the crankshaft, and for these reasons the fixed radial design has had a great and well deserved popularity. Its weakness lies in the large frontal area presented, and in the difficulty of working-in the radially arranged cylinders, enlarged as they are by cooling fins, into a neatly streamlined fuselage form. The type suffers also from the severity of the loads imposed on the crankpin by the rotating and reciprocating masses at high speeds. The arrangement whereby the working effort of so many as nine pistons can be concentrated upon one crank and crankpin is very economical in crankshaft weight, but this economy has to be paid for by the concentration also of the inertia forces, which increase in proportion to the square of the speed and produce extremely severe loads upon the crankpin.

A review of those aero-engine types which have been successfully developed shows that cylinder sizes range from 3½ to 6 in. in diameter and from about 3½ to 7 in. in length of stroke. There is no marked difference in the range of sizes between the different types of air-cooled and water-cooled engines, or of 'in-line' and radial.

The real governing factors in this matter of cylinder size are the need to go to high speeds and to avoid overheating of the piston, on

the one hand, both of which make for the use of small cylinders; and a desire to avoid the excessive mechanical complication of a great number of small cylinders, on the other.

For reasons which will be clear in due course a greater horse-power per unit of volume swept through by the piston can always be obtained in a small, than in a large, cylinder; so that the greatest concentration of power for a given overall size of engine should be obtained from a large number of small cylinders. An aero-engine, however, has to provide a certain aggregate horse-power, and for the larger sizes of power unit, from 400 to 800 B.H.P., the complexity of the design and the cost of production of an engine with the necessary number of small cylinders, each complete with its ignition and valve-operating gear, would be most formidable.

Although the essentials of the engine are similar, there are certain special conditions governing its design and operation which set the aero-engine apart from others of its species. Apart from lightness, the most important special condition is that the engine cannot logically be considered apart from the airscrew which it drives. The efficiency with which the fuel supply is used in propelling an aeroplane is the product of the effective airscrew and engine efficiencies, and the conditions which have to be met to get the best airscrew efficiency affect engine design, and weight, so vitally, that any consideration of the power plant must be carried right through to the 'thrust horse-power' provided by the airscrew. An outline of aero-dynamic and airscrew theory in so far as these affect engine performance will be given in the next chapter.

Other special conditions affecting engines in the air are, of course, the changing pressure and temperature of the air at different altitudes. These have a direct effect upon the power output of a normally aspirated engine by their effect upon the weight of the fuel and air charge per cycle; and no less important indirect effects upon engine operation, because of the way in which the mechanical losses, cylinder cooling, and tendency to detonate are affected.

For maximum engine power combined with high airscrew efficiency some form of speed reduction gear is necessary between the engine crankshaft and the airscrew. The maximum safe rotation rate of the crankshaft is fixed by mechanical considerations, concerned mainly with the loads on the bearings and inertia forces due to the moving parts. With the perfection of modern designs these permissible rotation rates are very high, 2,500–3,500 r.p.m., and much higher than the rotational speeds for which it is possible to design an efficient airscrew for the usual conditions of working. In order to get the maximum power from an engine it must be

able to run at its maximum permissible crankshaft speed, and the most appropriate reduction gear, to link this crankshaft speed with the speed which the airscrew requires, will depend upon the design of the aeroplane in which the engine is carried, and its forward speed.

An engine would only occasionally be called upon to run at its maximum permissible speed. It would have, besides, a defined 'normal full speed' at which its 'rated full power' would be measured. The rated power would then be the B.H.P. which the engine was capable of giving at its normal full speed when tested under carefully defined conditions on the test bench. When an engine is provided with a supercharger for maintaining its ground-level power, the rated full power is usually specified as that which it will deliver at a certain height (the rated altitude) above ground-level; and this power is determined from tests at ground-level, to which corrections are applied in the manner explained in art. 76 of Chapter XIII. An engine which passes an official Air Ministry 'type test', according to which its rating is fixed, must run for 100 hours without a major failure of any kind, divided into ten 10-hour periods; the last hour of each period being run at the rated full power and the previous 9 at nine-tenths of this.

A considerable part of the working life of an engine may be expected to be spent under 'cruising' conditions. The engine would then be slightly throttled and would be running at rather less than nine-tenths of its normal full speed and delivering only about two-thirds of its rated full power. The relation between engine speed and power, in the air, is fixed primarily by the forward speed of the aeroplane, and by whether the latter is flying level, climbing, or losing height. During level flight the power delivered by the engine would be proportional to a power of the revolution speed between the 2·5th and the cube. The cube law is found to hold only near the top speed.

When an aeroplane increases its height the B.H.P. of its engine, if this is normally aspirated, will fall off at a rate depending upon a variety of factors. These factors will be examined in detail in the ensuing chapters, and the reasons will be explained which underlie the observed law of power variation with height. One fact of wide-spread influence may be mentioned at this stage, however, and that is the increased importance of the mechanical losses in the engine, and their affect upon the B.H.P. The lost horse-power (L.H.P.) at ground-level would not be more than about 10 per cent. of the full throttle I.H.P., but it might be 20 per cent. in an unsupercharged engine at 15,000 ft., although the actual horse-power absorbed by

friction and pumping loss shows a small decrease at altitude, as compared with ground-level conditions.

As an aeroplane rises the decrease of air density will, of course, affect the aero-dynamic conditions of flight in parallel, so to speak, with its effect upon the available engine power; and the flight conditions will themselves react upon the engine power through their influence upon the revolution rate of the airscrew and engine.

It should be kept constantly in mind that the r.p.m. of an engine in flight will depend upon two independent variables, namely, the forward speed of the aeroplane and the position of the engine throttle. These two are independent because the forward speed of the aeroplane is settled purely by the position of the elevator; it is affected scarcely at all by a change of throttle setting. To alter the engine throttle without any movement of the elevator will not produce a change of the forward speed, but a change in the inclination of the flight path, e.g. from level flight to climbing or losing height.

There will be a certain forward speed at every height which enables the aeroplane to climb at its maximum rate for a given throttle setting. It is found in practice that the effects of the falling air density upon the wings and airscrew, on the one hand, and upon the effective gas pressure in the engine cylinders, on the other, is to produce the best rate of climb with a normally aspirated engine at nearly the same forward speed and engine r.p.m. at all heights; a condition that corresponds to an air-speed indicator (A.S.I.) reading (see Ch. II) which decreases at the rate of about 1 m.p.h. per 1,000 ft. With a supercharged engine, on the other hand, the elevator must be adjusted so as to maintain a constant A.S.I. reading below the rated altitude, and in these circumstances the true speed and the engine r.p.m. will both increase nearly in proportion to $\dfrac{1}{\sqrt{(\text{air density})}}$.

ART. 2. *Methods of light cylinder construction.*

To turn now to the matter of cylinder construction. This is an aspect of aero-engine design which is both peculiar to the species and of great importance in connexion with heat-flow and cooling problems. It is proposed, therefore, to review some of the special methods which have been evolved, and this will serve to bring out the fact that the phenomenal reduction of weight per horse-power already noted has not been achieved by any mere cutting out of superfluous material.

From every point of view except those of extreme lightness and high thermal conductivity, cast iron is ideal as a material for cylinder

construction. It is cheap, close-grained, easy to cast and easy to machine, and it forms an admirable working surface for the piston. These virtues are sufficient to make its use universal for all automobile work, as well as for all the more slow-running engines of the power-house and marine types, whether employing steam or internal combustion.

Unfortunately its two weak points mentioned above are fatal when it comes to the design of an aero-engine, and the perfect substitute is not easy to find. Lightness can be got by making the cylinder barrels of thin steel tubes, but the valve gear on the cylinder heads requires rigidity under stress, and that means thick metal walls and a low thermal conductivity round the combustion space, just where the problem of getting rid of the waste heat rapidly is most acute.

Some of the ingenious and interesting methods of combining lightness and rigidity with good thermal conductivity are illustrated in figs. 1–4. In every type of cylinder, be it water-cooled or air-cooled, the governing consideration is that of providing an easy path of escape for the waste heat. The cylindrical working surface for the piston must be of steel, on grounds of strength; but the thermal conductivity of steel is no more than a quarter of that of aluminium, and round the cylinder head, where the heat-flow is mostly concentrated, steel cannot compete with the light metal in providing a ready path for the waste heat, combined with lightness and rigidity.

In the modern water-cooled engine, designed on the 'monoblock' system, rigidity is obtained by forming the heads of a row of cylinders in one common light alloy casting, through which the main water circulation takes place. Figs. 1 and 2 illustrate two alternative types of compound cylinder, composed of a steel liner with an aluminium head and water-jacket. In fig. 1 the top of the combustion space is formed by the closed end *A* of the steel liner, and in this steel top the seats for the valves are formed. The closed-ended liner is screwed into the aluminium casting *B*, and at the top of the liner the two form a metal-to-metal contact. In the design of fig. 2 the steel liner is open-ended. Again it is screwed into the head, at *C*, but in this design there is nothing but the aluminium wall between the combustion-space and the cooling water at *D*. Metal-to-metal contacts, however perfect when new, are apt to suffer by slight distortion and differential expansion in service, and the slightest failure of contact leads at once to overheating of the steel liner. In some early designs both of air-cooled and water-cooled cylinders the steel liner was screwed, shrunk, or cast, into an aluminium sheath, which carried the cooling fins on an air-cooled cylinder, or formed the inner wall of the water-jacket in a water-cooled cylinder. Such compound

walls, however, were never satisfactory in prolonged service, owing to the failure of good thermal contact at the metal-to-metal surface. To-day the almost universal practice is to use an open-ended steel barrel combined with an aluminium alloy head; and to have only sufficient overlap to provide the necessary strength of joint. The most usual method of making the joint between the barrel and the head in an air-cooled cylinder is described below.

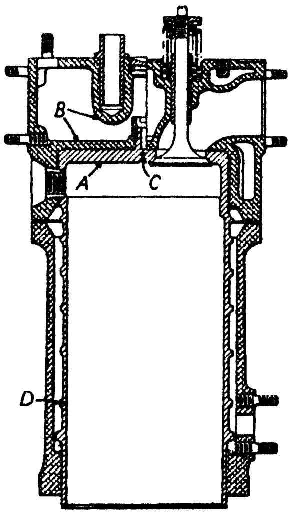


FIG. 1. Water-cooled cylinder with closed-ended steel liner.

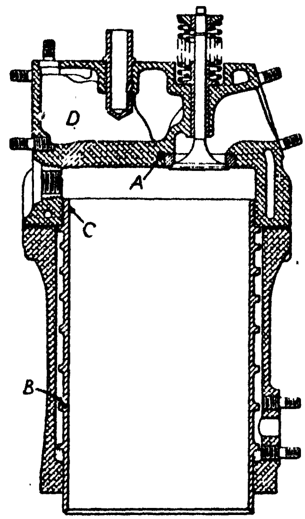


FIG. 2. Water-cooled cylinder with open-ended steel liner.

Another interesting form of water-cooled construction is that employed in certain of the Rolls-Royce designs, in which there is nothing but a compression joint, shown at *A* in fig. 3, between the steel liner and the cylinder head. Each bank of six open-ended steel liners are gripped between the cylinder-head and the crankcase castings, which are drawn together by long steel bolts, not shown in the illustration. The cylinder-head casting carries the outside wall of the water jacket *B*, which makes a second joint, with a seal of compressed rubber, against a rib towards the lower end of the cylinder liner, at *C*.

A section of a typical air-cooled cylinder is shown in fig. 4. The steel barrel is screwed into the aluminium alloy head, which carries at its lower end a single steel fin *A*, shrunk on, to lock the joint.

With any open-ended liner and aluminium head special valve-seats must be provided in hard metal, often steel or an aluminium-bronze, which are screwed or shrunk into the light alloy head while it

is hot, and finished when in position. The 'pent-house-roof' form of cylinder head, with overhead valves, shown in fig. 4, provides almost the ideal shape of combustion space so far as compactness and small surface-volume ratio are concerned. A good shape of combustion space is of vital importance in a high efficiency engine, for upon it depend freedom from detonation and the ability to use a high compression ratio.

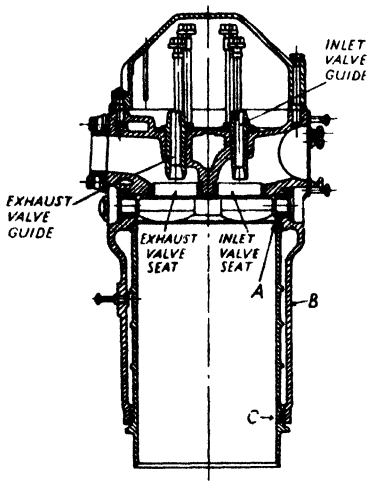


FIG. 3. Water-cooled cylinder (Rolls-Royce) with open-ended liner.

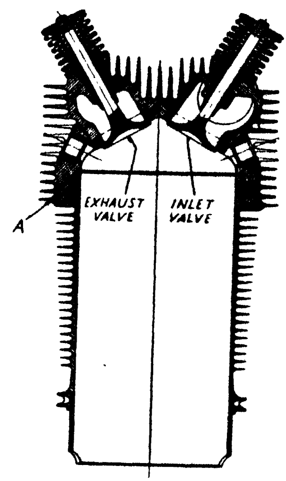


FIG. 4. Air-cooled cylinder with an open-ended steel barrel screwed into an aluminium alloy head.

ART. 3. *The standard atmosphere.*

In order to give a complete account of what an aeroplane can do, its 'performance' in terms of its maximum level speed and rate of climb must be stated for a number of different heights above ground-level.

Since the pressure, temperature, and therefore density of the air at any particular height above the ground may vary from day to day, it follows that the speed and rate of climb of the same aeroplane will vary from day to day at the same height; and in order to give any precise meaning to its 'performance' it is necessary to reduce the actual performance measured on a particular day to certain standard conditions. For this purpose an 'International Standard Atmosphere' has been defined in which the pressure, temperature, and density are given certain arbitrary fixed values for each 1,000 feet above sea-level; these values representing an average of the actual values throughout the year.

The temperature is taken as that given by the linear equation

$$T = 288 - 1{\cdot}9812h$$

in which h is the height measured in thousands of feet. For most practical purposes it suffices to take the fall of temperature as 2° C. per 1,000 ft. No simple equation expresses the relation between the standard pressure and density and the height, but figures are given in table 1 for the temperatures, densities, and pressures in the standard atmosphere up to 35,000 ft., above which the stratosphere at a uniform temperature of −55° C. is assumed to begin.[1*] The actual pressures and densities are not given in table 1, but only the

TABLE 1

The standard atmosphere.

Standard height feet	*Temperature ° C.*	*Relative temp. abs.* θ	*Relative density* σ	*Relative pressure* p
0	15	1	1·000	1·000
2,000	11·04	0·986	0·943	0·930
4,000	7·07	0·972	0·888	0·864
6,000	3·11	0·959	0·836	0·801
8,000	−0·85	0·945	0·786	0·743
10,000	−4·81	0·931	0·738	0·687
12,000	−8·77	0·917	0·693	0·633
14,000	−12·74	0·904	0·650	0·587
16,000	−16·70	0·890	0·609	0·542
18,000	−20·66	0·876	0·570	0·499
20,000	−24·62	0·862	0·533	0·459
22,000	−28·59	0·849	0·497	0·422
24,000	−32·55	0·835	0·464	0·387
26,000	−36·51	0·821	0·432	0·355
28,000	−40·47	0·807	0·402	0·325
30,000	−44·44	0·794	0·374	0·297
32,000	−48·40	0·780	0·347	0·271
34,000	−52·36	0·766	0·322	0·246
35,000	−54·34	0·759	0·310	0·235

values at each height relative to the standard ground-level conditions. In the great majority of aerodynamic calculations it is convenient to make use of these relative values. Whenever absolute values are required the appropriate value of p, σ, or θ has to be multiplied by the standard value at ground-level, namely

$$p_0 = 14{\cdot}7 \text{ lb. per sq. in.} = 2{,}117 \text{ lb. per sq. ft.}$$
$$\rho_0 = 0{\cdot}0765 \text{ lb. per cu. ft.}$$
$$= 0{\cdot}002378 \text{ slugs per cu. ft.}$$
$$\theta_0 = 288^\circ \text{ C. abs.}$$

* For all references in the text, see pp. xv and xvi.

The relative pressure, density, and absolute temperature at any height in the standard atmosphere are connected together by the equations

$$\theta = \sigma^{0 \cdot 235} \qquad p = \sigma^{1 \cdot 235}$$

$$\theta = p^{0 \cdot 19} \qquad \sigma = p^{0 \cdot 81}.$$

These relationships are important when we come to consider the law of variation of the power of an engine at different heights in the standard atmosphere.

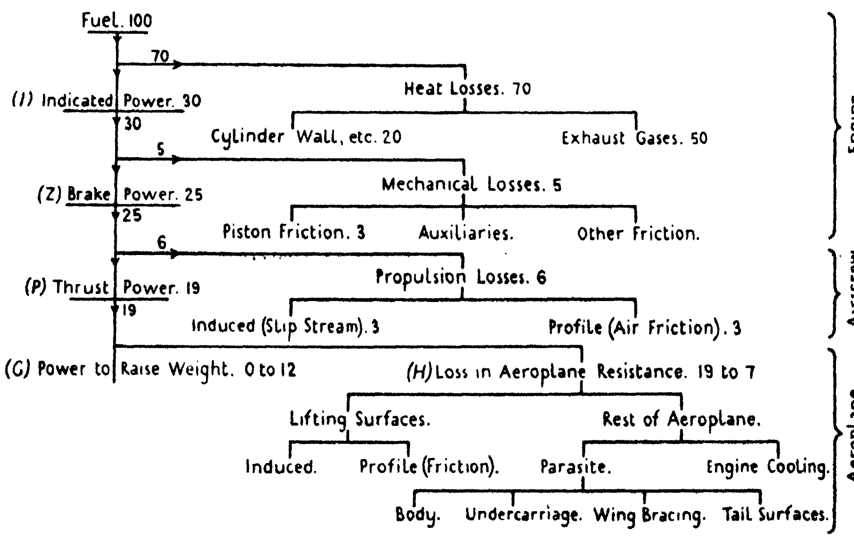
II

THE AEROPLANE AND ITS POWER PLANT

ART. 4. *The power balance sheet of an aeroplane.*

The propulsion of an aeroplane involves a series of processes in which the energy potentially present in the fuel is first realized as heat and is then converted into a form in which it can be used for overcoming the resistance of the air, or for raising the weight of the aeroplane against the force of gravity. In each process a part of the original store of energy is lost, and it is useful to have in mind the relative amounts received, transmitted, and rejected at each stage, because it presents in a proper perspective the various problems inherent in the aeroplane and its power plant.

TABLE 2



In table 2 the complete transformation is represented diagrammatically. The numerical values, of course, are given only as typical approximations. The first stage, from fuel to indicated engine power, and the second, from indicated to brake power, will form the subject of the main part of this book. It is the object of the present chapter to explain in outline the third and fourth stages, from brake power by means of the airscrew to thrust power, which last is devoted either as a whole or in part to overcoming the resistance of the complete aeroplane. No attempt will be made to give proofs of the relations which

govern the various aerodynamic losses, other than brief explanations from a physical standpoint. For fuller treatment the reader is referred to standard text-books on aeroplane mechanics.

It is useful to note at the outset that of each 100 units of energy present in the fuel some 25 only are, as it were, handed over to the aeronautical engineer. The majority of the remaining 75 are dissipated in losses inherent in the thermal processes characteristic of the engine, which have been dealt with in vol. i of this book. The greater part of this waste heat is disposed of without difficulty in the exhaust, but the remainder, equivalent to about two-thirds of the indicated power, must be removed from the walls of the cylinders and other parts in order that the fabric of the engine may survive. Whether or no some intermediate substance is interposed between the cylinder walls and the airstream, into which this heat must eventually go, its transfer to the air essentially involves a loss in air friction and generally also in turbulence. Owing to technical reasons which will be discussed in Chapters VI and VII these losses vary appreciably with the arrangement adopted, but they constitute an aerodynamic as well as a practical problem whose importance increases as the aeroplane itself is improved. On the scale adopted in table 2 the disposal of the 20 units from the cylinder walls (to which must be added the 3 units ascribed to piston friction) involves an aerodynamic loss which will be found in tables 3 and 4 (p. 17) under 'Aeroplane Resistance' as 'Engine Cooling'. This loss is not in practice less than 2, and may be as much as 4 units, a far from negligible proportion of the 19 units of energy which the airscrew transmits in the form of thrust.

The distribution of the thrust power P between that part which is used in raising the weight of the aeroplane, G, and that used in propelling it against air resistance, H, depends upon the condition of flight. In level flight $G = 0$ and $H = P$. When an aeroplane is climbing at its maximum rate, G is generally between $\frac{1}{3}$ and $\frac{2}{3}P$. It must be noted that G has no relation to the power used in generating the force which sustains the weight of the machine. The latter is represented by the 'induced' loss due to the lifting surfaces, a term which will be explained below.

ART. 5. *Wing lift—control of attitude and speed.*

The 'lift' L on an aeroplane is defined as the component of the total air reaction on it at right angles to the flight path. We are concerned only with conditions when the flight path is straight, but it may be inclined either upwards or downwards in relation to the horizontal. In practice, however, the angle of inclination, θ, of the

flight path to the horizontal seldom exceeds 15°,* and it follows that the component of the weight W of the machine normal to the flight path never differs from W by more than 3 per cent., and to an order of accuracy which is ample for our purpose we may therefore write, in all conditions,

$$L = W. \tag{1}$$

Moreover, in considering the equilibrium of the forces (though not of the moments) we may disregard the lift of all parts except the wings. Thus we arrive at the conclusion that the lift of the wings of an aeroplane in normal flight (whether 'flying level', 'climbing', or 'gliding' within the limits of ordinary practice) is constant, to a first approximation. Now the lift of a wing is proportional to the density ρ of the air; to the square of the speed V; and to the angle of 'attack' α, defined as the inclination of a reference line on the side elevation of the wing to the direction of flight, so chosen that the lift is zero when $\alpha = 0$. Hence it follows that

$$\alpha \propto \frac{1}{\rho V^2}. \tag{2}$$

Since the wings are normally fixed in relation to the structure of the aeroplane, it follows that the inclination of the aeroplane to the flight path is inversely proportional to ρV^2.

All pressures generated by the movement of the aeroplane through the air are proportional to ρV^2, and it is current practice to use the difference between the pressure in an open tube facing the 'relative wind' (the 'pitot' pressure) and the general barometric (or 'static') pressure, which is numerically equal to $\frac{1}{2}\rho V^2$, to operate a pressure gauge, the air speed indicator, or A.S.I., whose scale is marked with the corresponding speed when the air density has its 'standard' or nominal average ground-level value ($\rho_0 = 0{\cdot}0765$ lb. per cu. ft. at 15° C. and 760 mm.). When, as of course is generally true, the air density is less than this value, the A.S.I. reading, V_i, is less than the true speed in the ratio of the square root of the 'relative' density, $\sigma = \rho/\rho_0$. Thus

$$\rho_0 V_i^2 = \rho V^2$$

or

$$V_i = \sigma^{\frac{1}{2}} V. \tag{3}$$

It follows that to each A.S.I. reading, V_i,† there corresponds one value, and one only, of the angle of attack α.

* This covers what may fairly be described as the normal flight of aeroplanes with not more than 140 B.H.P. per 1,000 lb. of total weight. For more highly powered machines the simplified analysis which follows would need some modifications, but the essential conclusions are unaffected. The same remark applies to very steep dives.

† Often described as the 'indicated air speed'.

The lift of a wing ceases to be proportional to α at values above about 15°, owing to the 'stalling' of the flow. This occurs at speeds which are below the range properly described as normal flight, but it is useful to note that the resistance of the wing rises rapidly at angles in excess of this value.

It may be remarked in passing that it follows from what has been said above that the control of the speed of an aeroplane, as mentioned in art. 1, is effected by changing its attitude, i.e. its angle of attack, and not, as for example with a road vehicle, by opening or closing the throttle of the engine. The latter affects only* the slope of the flight path θ. The attitude is controlled by means of the tail-plane or elevator, of which a change in the inclination relative to the wing applies to the machine a turning-moment in a vertical plane, and this rotates it from its original attitude into another in which the moments of the whole of the air-forces on the various surfaces round the centre of gravity are once again in equilibrium. If this adjustment is made fairly slowly the speed of the machine changes to correspond with the attitude, in accordance with the relations developed above, but it must be noted that if the throttle is untouched there will also be a change in the inclination θ of the flight path to the horizontal, since the power required to propel the machine varies with the speed. The inter-relation of speed, inclination of the flight path, and throttle setting is in fact more complex than a superficial examination would suggest. A full treatment would require more space than is available here. It may be summarized formally, however, by saying that

(1) The inclination α of the machine to the flight path is substantially a function of the tail-plane (or elevator) setting only, the governing physical condition being a balance of the moments due to aerodynamic forces round the centre of gravity.

(2) The A.S.I. reading V_i is a function of α only, as pointed out above, and is conditioned physically by a balance of the forces normal to the flight path.

(3) The inclination θ of the flight path itself to the horizontal, on the other hand, is a function of the two *independent* quantities, namely speed of flight and throttle setting, which correspond to a balance of the forces along the flight path; or, from another point of view, to a balance of the energy supplied to, and required by, the machine.

It is the last of these three conditions in which the engine designer is chiefly interested.

* In practice it may affect the attitude indirectly on account of the change of the airscrew slip stream over the tail control surfaces.

ART. 6. *Drag—Power—Propulsive efficiency.*

The resistance or 'drag' D of an aeroplane is defined as the component of the air reaction upon it along the flight path, in a backward direction. The airscrew thrust T is not included. The inclination of the airscrew axis to the flight path, which is in practice small and variable, may be neglected and it follows that we may write either (*a*) by consideration of the balance of forces along the flight path, assumed to be inclined at the (small) angle θ to the horizontal,

$$T = D + W\theta, \tag{4}$$

or (*b*) by consideration of the balance of power supplied and required,

$$P = H + G, \tag{5}$$

where P = the thrust power of the airscrew $= TV$ ft.-lb. per sec.

H = drag power $= DV$ „

and $G = Wv_c$ $= WV\theta$ „

v_c being the rate at which the machine gains height, or the 'rate of climb'. Equation (5) is obviously equation (4) multiplied throughout by V.

The thrust power of the airscrew, P, is given by

$$P = \eta Z, \tag{6}$$

where Z is the B.H.P. of the engine and η is the airscrew efficiency. The latter quantity will be dealt with more fully in art. 11. A representative average value for the airscrew efficiency, 76 per cent., has been given in table 2. It may be noted here, however, that the quantities T, P, and η differ appreciably from their values for the 'free' airscrew (i.e. when isolated from the aeroplane), and include an allowance for the effect of the airscrew 'wash' or 'slip stream' in increasing the drag of the parts of the aeroplane within it. Experiment shows* that this effect may be taken into account completely by deducting from the thrust or thrust power of the 'free' airscrew a certain fraction, which in practice varies from 4 to 12 per cent. according to the arrangement of the parts, but is constant for a given aeroplane and airscrew. It follows that the effective or 'propulsive' efficiency η, is on an average some 92 per cent. of that of the isolated airscrew. This apparently artificial method of dealing with the problem has not only the merit of simplicity, but also that of making it possible to regard the aeroplane and the airscrew as to some extent separate bodies. Their mutual interaction cannot be treated satisfactorily from the obvious starting-point of the actual tension or

* This result is also suggested by theoretical considerations.

thrust in the link which in fact connects them, i.e. the airscrew shaft. This force generally exceeds not only the 'propulsive thrust' T but also that of the isolated screw, by reason of the existence of a region of high pressure between the screw and the nose of the body supporting it, which involves, however, no direct loss of energy. The loss of energy arises mainly from the higher speed of the slip stream, but sometimes also from an unfavourable effect on the form of the flow round the bodies in it.

ART. 7. *Wing drag. Induced and profile drag.*

The contributions of the various parts of the aeroplane to the total drag D and the corresponding power H are shown diagrammatically for two typical conditions of flight in tables 3 and 4. The

TABLE 3

Subdivision of resistance when cruising.

- Total resistance at cruising speed. 100
  - Lifting surface. 35
    - Induced. 7
    - Profile. 28
  - Rest of aeroplane. 65
    - Parasite. 50
      - Body. 15
      - Undercarriage. 20
      - Wing bracing. 8
      - Tail, etc. 7
    - Engine cooling. 15

TABLE 4

Subdivision of resistance when climbing.

- Total resistance at climbing speed. 100
  - Lifting surface. 60
    - Induced. 50
    - Profile. 10
  - Rest of aeroplane. 40
    - Parasite. 30
      - Body. 9
      - Undercarriage. 12
      - Wing bracing. 5
      - Tail, etc., 4
    - Engine cooling. 10

Note.—During the last few years there has been a marked tendency towards a reduction of parasite resistance in comparison with that due to other causes. Figures as low as one-half those quoted above have been attained.

manner in which the various contributions depend upon the chief proportions of the machine will now be summarized, at first dealing

with the wings only, and in the next article with the complete aeroplane.

The drag of the wings arises in two distinct ways. The first part, the 'induced drag', is due to the energy left in the air on account of the disturbance essential to the production of lift. The second part, the 'profile drag', is due primarily to the skin friction of the air moving over the exposed surfaces. The induced drag D_i is directly proportional to the square of the lift, and inversely proportional to the square of the span B of the wings; to the air density; and to the square of the speed. It is independent of the 'profile', or side elevation of the wing. Hence we may write

$$D_i = a\frac{L^2}{\rho V^2 B^2}. \tag{7}$$

The profile drag D_0 is directly proportional to the area of the wing surface S; to the air density; and to the square of the speed. By contrast, it is not directly affected by the lift, but depends on the shape of the profile. Thus,

$$D_0 = b\rho V^2 S. \tag{8}$$

The quantities a and b are mere numbers, since ρV^2 is of the nature of a pressure per unit area. Theory suggests that a should be least for a wing so shaped that the distribution of lift across the span follows the ordinates of a circle, for which it has the value $2/\pi$. Normal wings have a more nearly uniform lift distribution, but a is seldom more than 5 per cent. greater than $2/\pi$. For a biplane of the proportions usually adopted these values are reduced about 20 per cent.

The quantity b must be found experimentally, but for a good 'streamline' wing profile it is approximately equal to the 'skin friction coefficient' for a thin flat plate, on the same basis, namely 0·004. To an appreciable extent b depends on the angle of attack of the wing, owing to the fact that the flow follows the profile most closely at one particular angle. For our purpose this may be taken into account by using an average value, in practice from 0·005 to 0·006 for good profiles, over the whole working range.

As the wing stalls, b increases rapidly. a rises also to some extent, since the centre part of the wing stalls at a lower angle of attack than the outer parts, so that the lift per unit span tends to fall at the centre and to rise towards the tips. These effects are, however, outside our present scope.

It may be noted that in order to calculate forces from relations such as equations (7) and (8) it is necessary to use a unit of mass such that force = mass × acceleration. Thus if, as usual, the unit of force

is the lb. weight, the unit of mass on the ft.-sec. system is the slug (32·2 lb.). The standard air density ρ_0 has been given on p. 10 as 0·002378 slugs per cu. ft.

The division of the resistance, and of the corresponding power-loss, associated with any body which generates a reaction in a fluid, into 'profile' and 'induced' parts is of such wide interest that an explanation in general terms of the underlying mechanism is called for.

Profile resistance arises directly from the action of the viscosity of the fluid. This action is most intense near the surface of the body where large rates of shear (or velocity gradients) exist. The formation of a 'boundary layer' of retarded fluid, 'skin friction', and in extreme cases, 'breakaway' of the flow from the surface, and 'turbulence', are its direct consequences. The evidence of its existence in the fluid lies in the 'wake' of eddies immediately behind the body. Elsewhere no trace remains of the passage of a streamlined body, the surrounding fluid having yielded to it by a gentle outward and return motion involving no appreciable loss of energy. The circumstances of the flow and the associated force 'coefficients' (*force per unit area* $\div \rho V^2$) depend upon the Reynolds Number, i.e. on the relative parts played in the balance of actions and reactions by inertia and viscosity. In the conditions with which we are here* concerned experiment shows that the force coefficients are substantially constant for a given attitude of the body; so that the forces, as stated above (eqn. (8)) are proportional to ρV^2 and to the 'surface', or more generally the square of the scale of the body. Profile resistance diminishes with a fall of the viscosity of the fluid (i.e. with a rise of Reynolds Number) and becomes zero for the ideal fluid of classical hydrodynamics in which all flows are streamline.

Induced resistance, by contrast, has no direct relation to viscosity, but is due to the continual supply of energy needed to produce the general disturbance of the air without which the reaction in question (in this case the lift) could not exist. This disturbance is substantially confined to the region behind the wing. It consists of a downward motion directly behind it, and an upward motion on either side, and corresponding transverse motions above and below. It is most intense in regions nearest to the wing and to the path 'swept' by it, but no definite limit can be set to the air involved. It is accompanied by pressure variations which ultimately convey the lift on the wing to the ground below in the form of a travelling pressure wave, which, it is perhaps hardly necessary to say, is of extremely small intensity.

* This does not cover the problem of heat transfer as it arises in engine cooling (Chs. VI and VII).

The rate of supply of the momentum involved is proportional to the lift L; that of the kinetic energy is proportional to the power required for propulsion, i.e. to the resistance D multiplied by the speed V. The following argument, while not entirely beyond criticism, expresses the relation of these quantities compactly. The momentum generated per second is proportional

(1) to the density of the fluid ρ,
(2) to the square of a typical linear transverse dimension of the flow pattern, e.g. the span of the wing B,
(3) to the distance covered per unit time in the direction of flight, i.e. to the flight speed V, and
(4) to a typical velocity of the transverse disturbance, say v.

Therefore $$L \propto \rho B^2 V v.$$

Similarly, the kinetic energy generated per second is proportional to the same first three quantities, but to v^2, and therefore

$$DV \propto \rho B^2 V v^2.$$

Eliminating v, we have

$$D \propto \frac{L^2}{\rho B^2 V^2}. \quad \text{Cf. eqn. (7).}$$

The induced flow persists for a long time after the aeroplane has passed, since it involves no large rates of shear and hence is but slowly damped out by viscosity. It may readily be observed, particularly where it is most intense (immediately behind the wing tips), by flying through a thin 'smoke curtain'.

ART. 8. *The drag of a complete aeroplane.*

The resistance of the rest of the aeroplane, or the 'parasite drag', D_p (in which we may here include that due to engine cooling, see tables 2, 3, and 4), depends on the shape and size of its component parts, and is proportional to ρV^2. A value representative of a typical modern aeroplane is 5 per cent. of the weight of the machine, at 100 m.p.h.*

We may express the two parts of the wing drag discussed in the

* As mentioned earlier in the note to tables 3 and 4, values as low as $2\frac{1}{2}$ per cent. have been attained recently, by eliminating the undercarriage (by retraction) and external wing bracing, and by improving the shape of bodies and engine nacelles.

last article in a similar way, as a percentage of the weight, for, dividing (7) and (8) throughout by W (or L), we have

$$\frac{D_i}{W} = a\frac{1}{\rho V^2}\frac{W}{B^2} \tag{7 a}$$

and

$$\frac{D_0}{W} = \frac{b\rho V^2}{W/S}. \tag{8 a}$$

The quantity $W/B^2 = \overline{w}$ is known as the 'span loading', and $W/S = w$ as the 'surface loading'. The former is usually between 2 and 3 lb. per sq. ft. and should clearly be as low as general considerations of structural weight and convenience allow. The latter varies between 8 and 20 lb. per sq. ft., a low value being desirable in order to produce a low 'stalling' speed but clearly undesirable from the point of view of resistance. We may take $\overline{w} = 2{\cdot}5$ and $w = 15$* as representative values; and for a and b 0·67 and 0·005 respectively. Finally, if for ρV^2 we write, from equation (3), $\rho_0 V_i^2$ and, further, take 100 m.p.h. as the unit of speed, we have for the total drag D

$$\left.\begin{aligned}\frac{D}{W} &= \frac{D_i}{W}+\frac{D_0}{W}+\frac{D_p}{W}\\ &= \frac{0{\cdot}033}{V_i^2}+0{\cdot}017V_i^2+0{\cdot}050V_i^2.\end{aligned}\right\} \tag{9}$$

This expression must not be used at speeds much below that at which the lift ceases to be proportional to the angle of attack. For wing profiles now generally used this occurs when the lift L divided by $\rho V^2 S$ (i.e. the 'lift coefficient'), is about 0·45, corresponding to an angle of attack of about 12°. Thus V_i must not be less than the value given by $W/\rho V^2 S = 0{\cdot}45$ or, again using 100 m.p.h. as the unit, by

$$V_i^2 = \frac{w}{23}.$$

With a surface loading of 15 lb. per sq. ft., equation (9) can be safely used down to a value of $V_i^2 = 0{\cdot}65$ (i.e. 80 m.p.h.). It may be noted that this speed is about 20 per cent. in excess of the 'stalling speed' of the machine, which will correspond to the maximum value of the lift coefficient, generally about 0·65.

It is instructive to tabulate the results of substituting a series of values of V_i in equation (9). It is convenient to take 1,000 lb. as the unit of weight, thus giving the drag per 1,000 lb. of total weight as

* It is significant of the trend of aircraft development that fifteen years ago one would have regarded a surface loading of 10 lb. per sq. ft. as high.

set out in table 5 and fig. 5. The diagram emphasizes a conclusion which may readily be deduced from the form of equation (9), that the

TABLE 5

Drag per 1,000 lb. of total weight.

$\bar{w} = 2{\cdot}5$. $w = 15$. $D_p/W = 0{\cdot}050$ at 100 m.p.h.

V_i *m.p.h.*	*Induced*	*Profile*	*Parasite*	*Profile and Parasite*	*Total*
80	52	11	32	43	95
100	33	17	50	67	100
120	23	24	72	96	119
140	17	33	98	131	148
160	13	43	128	171	184
180	10	55	162	217	227
200	8	68	200	268	276

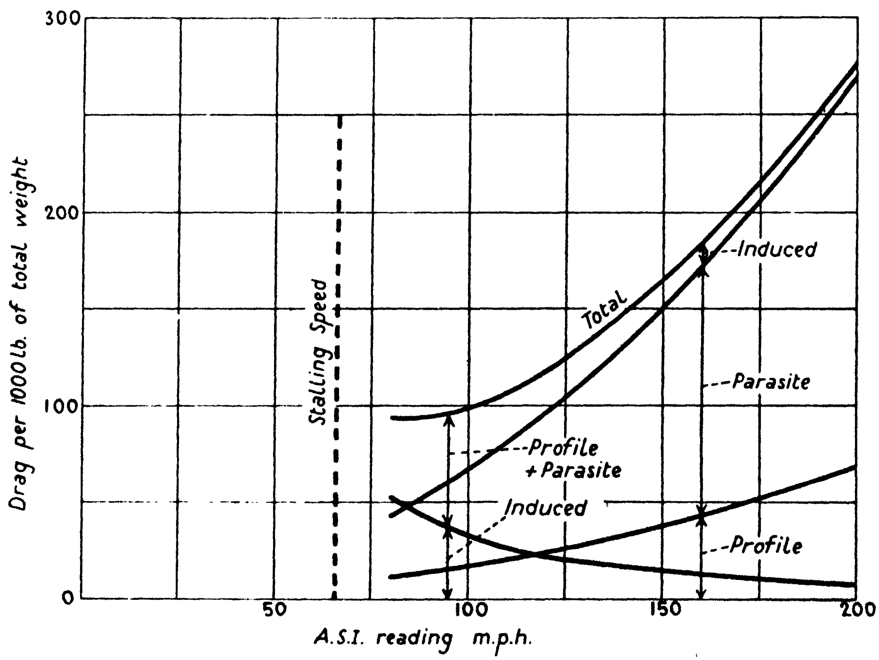


FIG. 5. Drag per 1,000 lb. total weight for a typical aeroplane.

total drag has a minimum value when the induced drag is equal to the sum of the profile and parasite drags. In this case the minimum occurs at 84 m.p.h., when the total drag is 94 lb. per 1,000 lb. total weight. As the speed rises the 'V_i^2' drag rapidly becomes the

predominating part, being 90 per cent. of the whole at 150 m.p.h. and 97 per cent. at 200 m.p.h., a range which covers the 'cruising' speed and 'top' speed of a machine of this type.

The form of equation (9) emphasizes the important generalization that the drag of an aeroplane of a given weight depends only upon the A.S.I. reading, V_i (i.e. upon the angle of attack α). The speed is found by dividing V_i by the square root of the relative density σ corresponding to the particular height at which the aeroplane is flying, whatever that may be.

ART. 9. *The thrust power required to propel the aeroplane.*

To express the 'thrust power' H on similar lines we must multiply D by V, and it follows from the last article that H does not depend on V_i only, but also on the air density. At any given V_i, H is inversely proportional to the square root of σ, the relative density. We may conveniently calculate the thrust horse-power per unit of weight from equation (9) by multiplying by V in 100's of m.p.h. and by $\frac{147}{550} = 0{\cdot}267$. If we write $V_i/\sigma^{\frac{1}{2}}$ for V, equation (9) leads to

$$\frac{H}{W} = \frac{1}{\sigma^{\frac{1}{2}}}\left[\frac{0{\cdot}0088}{V_i} + 0{\cdot}0044V_i^3 + 0{\cdot}0134V_i^3\right]. \qquad (10)$$

Thus at any given weight the power required for induced losses is inversely proportional to the speed, that for profile or parasite losses directly proportional to the cube of the speed. In table 6 and fig. 6

TABLE 6

Thrust horse-power required per 1,000 *lb. of total weight* (*at ground-level*).

$\bar{w} = 2{\cdot}5$. $w = 15$. $D_p/W = 0{\cdot}050$ at 100 m.p.h.

V_i *m.p.h.*	*Induced*	*Profile*	*Parasite*	*Profile and Parasite*	*Total*
80	11·0	2·2	6·8	9·0	20·0
100	8·8	4·4	13·4	18·8	27·6
120	7·4	7·6	23·1	30·7	38·1
140	6·2	12·1	36·9	49·0	55·2
160	5·4	18·0	54·8	72·8	78·2
180	4·8	25·6	78·0	103·6	108·4
200	4·4	35·2	107·0	142·2	146·6

the contributions of the various parts are shown, again per 1,000 lb. of total weight for conditions at ground-level. The corresponding

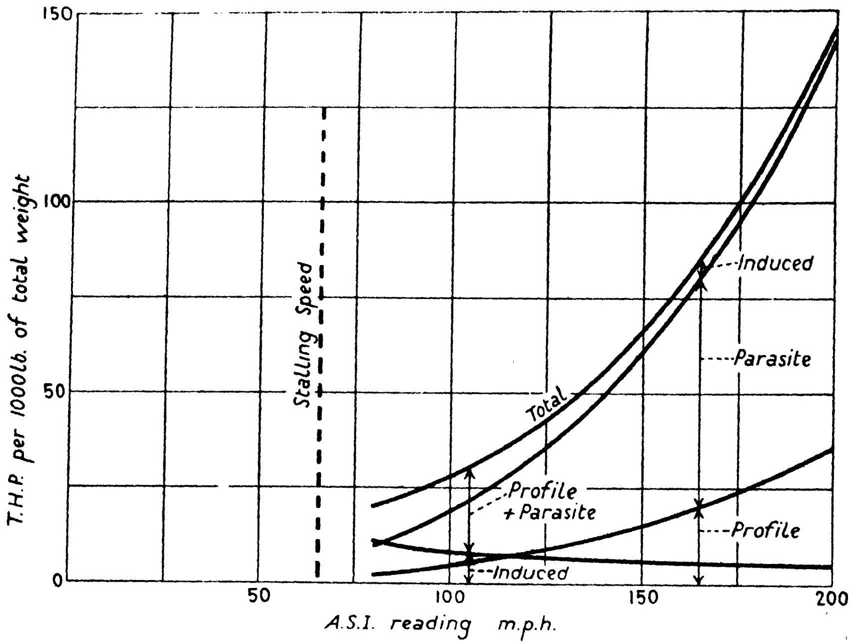


FIG. 6. Thrust horse-power per 1,000 lb. total weight for a typical aeroplane.

quantities and the speed at any other height are obtained by multiplying throughout by $1/\sigma^{\frac{1}{2}}$, values of which are given in table 7. These

TABLE 7

Average air density at various heights.

Height ft.	Relative density σ	$\frac{1}{\sqrt{\sigma}}$	$\frac{1}{\sqrt{\sigma}} \div \sigma$
0	1	1	1
10,000	0·738	1·17	1·6
20,000	0·533	1·37	2·6
30,000	0·374	1·64	4·4
40,000	0·245	2·02	8·2
50,000	0·152	2·56	17·0

tables contain the essence of the aerodynamic properties of the aeroplane which are of direct concern to the engine designer.

ART. 10. *Minimum power required for flight—The effect of air density —The attainment of high speeds.*

It may be noted that although the form of equation (10) implies the existence of a speed at which the power required for flight is a minimum, the speed in question is below the range within which

the formula applies with the necessary accuracy, as is clear from fig. 6. The true minimum power is not, however, appreciably different from that at the lowest speed quoted in table 6. Moreover it is at or about this speed that the aeroplane's rate of climb is greatest, and it is at this A.S.I. reading that it flies at its 'ceiling'. At the ceiling, however, the true speed V, and the power required, are both greater than the values in table 6 (80 m.p.h. and 20 h.p. per 1,000 lb.) in the ratio of $1/\sigma^{\frac{1}{2}}$, taken at the appropriate height from table 7. Thus at 30,000 ft. they become 130 m.p.h. and 33 h.p. per 1,000 lb. It will be seen that more than half the minimum power required for flight is due to the 'induced' loss, and it follows from eqn. (7a) that for a high rate of climb and a high ceiling it is desirable to have a low span loading, i.e. a large span.

Moreover it will appear later (Ch. XII) that the maximum power developed by an engine (if it is not supercharged) falls off with height rather more rapidly than the air density. Thus column 4 of table 7, which is the ratio of columns 3 and 2 $\left(\frac{1}{\sigma^{\frac{1}{2}}} \div \sigma = \frac{1}{\sigma^{\frac{3}{2}}}\right)$ gives a slightly conservative estimate of the ratio which the maximum power of the engine at ground-level must bear to the minimum (brake) power required for flight at ground-level, in order that the aeroplane may be able to fly level at the height shown in column 1. The very rapid rise of this ratio at great heights is reflected in the history of the height record, which, after rising from practically zero to 20,000 ft. between 1910 and 1914, reached 30,000 ft. in 1918, 40,000 ft. in 1929, and 47,000 ft. in 1934.

The attainment of high speeds will be seen from table 6 to involve (as in other forms of transport) a supply of power substantially in proportion to the cube of the speed. This obviously uneconomical requirement can be avoided only by reducing the profile and parasite drag; both by using high surface loadings (small wing surface), and, still more vital, by removal of the sources of parasite drag, i.e. by 'streamlining'. A top speed of the order of 200 m.p.h. is to-day reached by aeroplanes having the characteristics underlying table 6. It will be seen that such an aeroplane demands about 150 T.H.P., or 180 B.H.P., per 1,000 lb.; and that it weighs, therefore, $5\frac{1}{2}$ lb. per B.H.P. To drive the same machine at 400 m.p.h. would require eight times this power, that is to say nearly $1\frac{1}{2}$ B.H.P. per lb. of total weight of the aeroplane. In actual fact a speed approximating to this has been attained by an aeroplane having less than one-third of that h.p. weight ratio, namely about 450 B.H.P. per 1,000 lb., and it must have had, therefore, about one-third of the drag per unit weight of the aeroplane of table 6.

When high speed at great heights (and not at ground-level) is the objective, the reduction of parasite drag is still the outstanding requirement, but the power needed to attain a given speed with a given aeroplane is not simply proportional to the density of the air on account of the appreciable effect of induced drag. This can easily be seen by a simple manipulation of the figures of table 6. For example, 200 m.p.h. at about 22,000 ft. (where $\sigma = 0{\cdot}49$ and $1/\sigma^{\frac{1}{2}} = 1{\cdot}43$) means an A.S.I. reading of $\frac{200}{1\cdot43} = 140$ and hence a power of $55{\cdot}2 \times 1{\cdot}43 = 79$, which is 10 per cent. in excess of σ times the power needed for 200 m.p.h. at ground-level ($0{\cdot}49 \times 146{\cdot}6 = 72$). This is because, on account of the reduction of density, the induced loss rises from 3 to 11 per cent. of the total. The point is rather of qualitative than quantitative importance, as it indicates the way in which the proportions of an aeroplane which is to work at a great height should differ from those of a machine whose working height is only a few thousand feet.

ART. 11. *The airscrew. Equilibrium of engine and screw.*

While the engine designer can hardly fail to be interested in the characteristics of the aeroplane which determine the power required for flight, his concern with the airscrew, by which the brake power, Z, is transformed into the thrust power, P, is even more essential and intimate; for the direct mechanical connexion between the air-screw and the engine, nowadays usually in the form of reduction gearing, profoundly affects not only the operation but also the general layout and in many respects the detail design of the power plant.

From a purely aerodynamic standpoint the airscrew blades can be regarded as aerofoils (i.e. wings) of a somewhat complex form, whose motion is a combination of translation and rotation. Every point on the screw moves along a helical path, and for a given ratio of forward speed V to rotational speed n all the helices have one feature in common, namely their pitch, or advance per revolution, V/n. The ratio of this length to the diameter D of the screw, V/nD, clearly determines the pitch angles of all the helices, although the actual angle of each helix will depend upon its distance from the airscrew axis. Just as the lift and drag of a wing moving in a straight path are determined by its angle of attack, so the characteristics of an airscrew —torque, thrust, and efficiency—are determined by V/nD. The general form of the relations connecting these quantities with V/nD may be realized from the following argument.

If the rotational speed, n, is supposed to be maintained fixed (and the appropriate power supplied in any convenient way), the 'working condition' of the screw (or the pitch of the helices described by it)

can be varied by allowing the screw to advance at a series of speeds V from zero upwards. When $V = 0$ a large torque is required to rotate the screw, and a considerable thrust is developed, but no useful work is done, and the efficiency, η, is zero. As V rises the torque and thrust fall, owing to a reduction in the angle of attack of the blades to their effective helical paths, but the efficiency rises rapidly, since useful work (thrust $\times V$) is now being done. At a certain value of V, when the pitch of the motion is approximately equal to the geometrical 'pitch' of the blades, the thrust will fall to zero, the average angle of attack of the blades being also nearly zero. Here again the efficiency will be zero, and it follows that it will reach a maximum value at some intermediate value of V, or in general at some intermediate value of V/n for the particular screw, or of V/nD for all screws of the same shape. In practice the maximum efficiency occurs at a value of V/nD about three-quarters of that at which the thrust vanishes.

For a given screw both the torque and thrust at any given V/n are proportional to the density ρ and to the square of the rotational speed n. Hence the efficiency, which is equal to $\dfrac{\text{thrust} \times V}{2\pi \times \text{torque} \times n}$ depends upon V/n only.* Typical curves of $\dfrac{\text{torque}}{n^2}$ and efficiency against V/n are shown in figs. 7 A and 8 A. The precise forms of these curves depend somewhat on the shape of the screw, and for details of the calculation by which they are determined text-books on airscrew design must be consulted.

The primary condition of equilibrium of the screw and engine is

$$\text{engine torque}\dagger\ Q_E = \text{airscrew torque } Q_A. \tag{11}$$

This relation determines completely the advance per revolution at any forward speed in much the same way as the equality of the lift and weight of the aeroplane determines its angle of attack at any forward speed. For if the various controls which influence the power of the engine (throttle, ignition, mixture) are fixed, then in fixed atmospheric conditions its power and hence its torque depend only on the rotational speed n.

If now we rewrite equation (11) as $Q_E/n^2 = Q_A/n^2$, the left-hand side depends only on n and the right-hand side (since ρ is fixed), only on V/n. It follows that *for the chosen atmospheric conditions and settings of*

* It is perhaps worth while noting that the efficiency of a screw is entirely independent of the density of the air.

† Any gear reduction between the screw and engine is here considered as part of the engine.

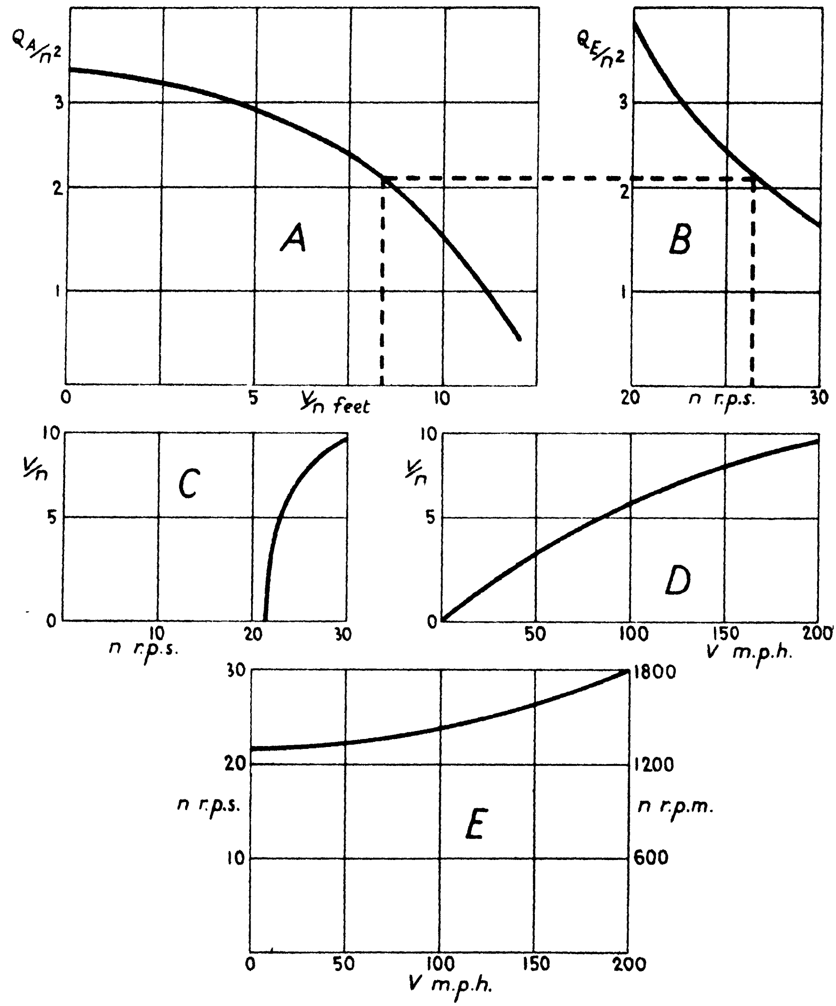


FIG. 7. Torque-speed relationship for airscrew and engine.

the engine controls there is a unique relation between n and V/n and hence one between n and V. In practice these relations must be found graphically, since the characteristics both of the screw and the engine cannot be expressed by any simple formulae. They are illustrated in a particular case by fig. 7.

In this diagram the engine is supposed to give a constant torque at all speeds, the variation of Q_E/n^2 with n being as shown in fig. 7B—in this case a simple curve of the form $y = 1/x^2$. The determination of the corresponding values of n and V/n is shown by the construc-

tion lines in figs. 7A and B. The results are shown in figs. 7C, D, and E.*

Now from what has been said above regarding the dependence of η upon V/n only, it follows that η is fixed at each value of V, and hence in general that *when the screw is coupled to a given engine, at a fixed setting of all controls and in given atmospheric condition, there is, for each forward speed, one value and one only of the rotational speed and the efficiency.*

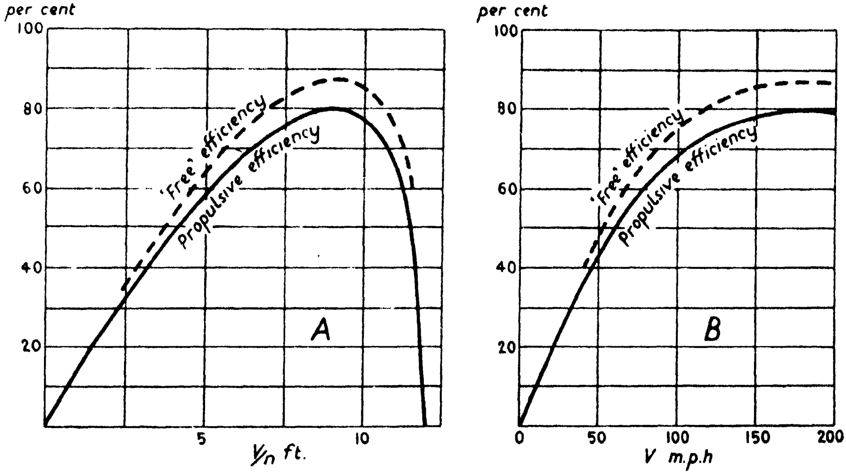


FIG. 8. Airscrew efficiency curves.

If fig. 8A represents the relation between η and V/n, then by use of fig. 7D we derive the relation between η and V (fig. 8B).

The two final diagrams, figs. 7E and 8B, constitute a complete answer to the problem of the equilibrium of the engine and airscrew, subject always to the above-mentioned restrictions as to the controls and the atmospheric conditions. A change in the setting of the engine controls is equivalent simply to substituting a different engine; and a different curve of Q_E/n^2, fig. 7B, would be required. A change of atmospheric conditions affects both Q_A/n^2 and Q_E/n^2, the former being reduced in proportion to the density ρ, the latter according to a more complex relation, which will be dealt with at length in Ch. XII. It may be noted, however, that in an unsupercharged engine Q_E, at a given throttle setting, is roughly proportional to ρ, and hence that figs. 7C, D, and E, and therefore fig. 8B, are nearly

* The numbers in these figs. refer to an engine giving 500 B.H.P. at 1,720 (airscrew) r.p.m. and a screw of 8·8 ft. diameter of such proportions that it absorbs this power when $V/n = 9{\cdot}2$ ft.

independent of atmospheric conditions. It follows that for an unsupercharged engine on full (or any fixed) throttle setting, the curves of rotational speed n and airscrew efficiency η against forward speed V are to a first approximation independent of atmospheric conditions.

ART. 12. *Performance of the complete aeroplane-screw-engine.*

The transition from figs. 7E and 8B to the curves representing thrust and thrust-power calls for no comment. The results are shown in fig. 9. For the combination to which figs. 7, 8, and 9 refer, speeds

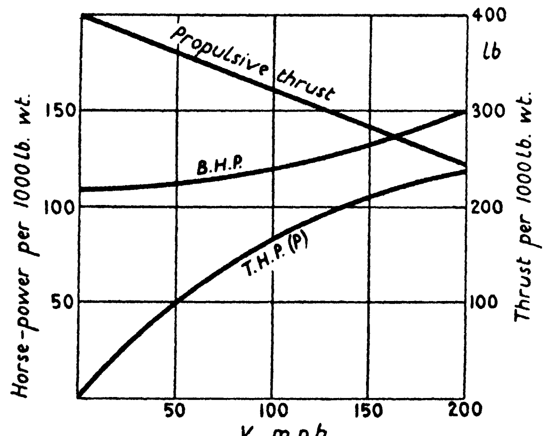


FIG. 9. The variation of propulsive thrust and H.P. with forward speed.

between zero and 75 m.p.h. would arise only during take-off. The thrust is then large, but it is important to notice that the calculations cannot be relied upon to give accurate results at speeds below say 55 m.p.h. On the whole the tendency is to over-estimate the thrust at lower speeds because the 'stalling' of the blades, which will persist in some degree to, say, 50 m.p.h., leads to losses which cannot be predicted.

It will be observed that the whole of the calculations outlined in the preceding section are independent of the aeroplane to which the engine and airscrew are fitted. The application of the results to the prediction of the performance of the aeroplane has already been outlined in art. 6, but it must be noted that, as there mentioned, the thrust power must first be reduced by a constant fraction (about 8 per cent.) to allow for the increased drag of parts of the machine in the 'slipstream' of the airscrew. Alternatively, the efficiency used above in figs. 8A and B may be the 'propulsive' efficiency (about 92 per cent. of the 'free' efficiency), thus giving the thrust power P of

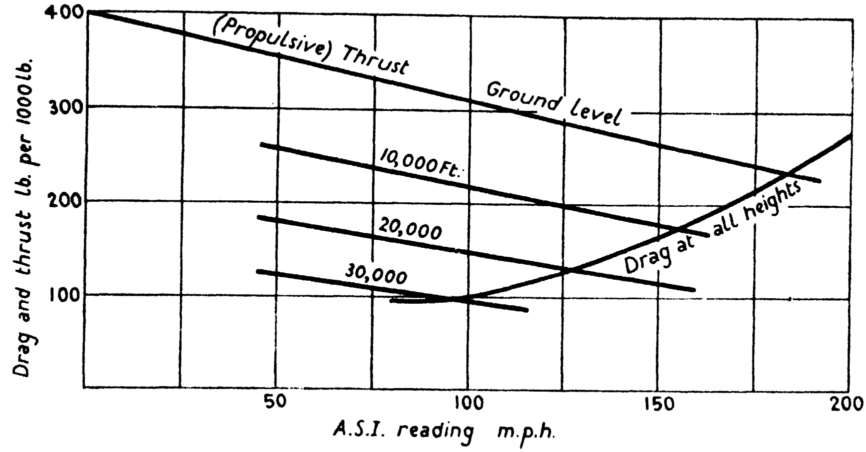


FIG. 10. The balance of drag and propulsive thrust.

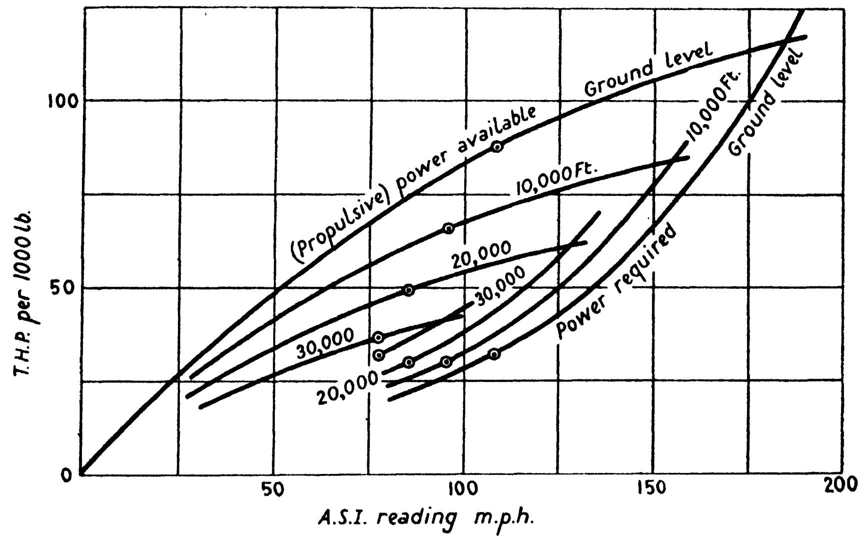


FIG. 11. The balance of 'power available' and 'power required'.

equations (5) and (6) in art. 6, and the corresponding 'propulsive thrust' T. In principle it is immaterial whether the final calculation of speed and rate of climb is made by consideration of the thrust and drag (eqn. (4)), or of the power (eqn. (5)). In practice the former is distinctly quicker, but the latter has the advantage that it retains more direct touch with the power of the engine. In figs. 10 and 11 the two processes are shown graphically, the aeroplane having the characteristics

of tables 5 and 6, and the maximum power of the engine at ground-level being such that the B.H.P. per 1,000 lb. is 140, which corresponds to a power loading of 7 lb. per B.H.P. The total weight for which the engine and airscrew of figs. 7 and 8 are appropriate is

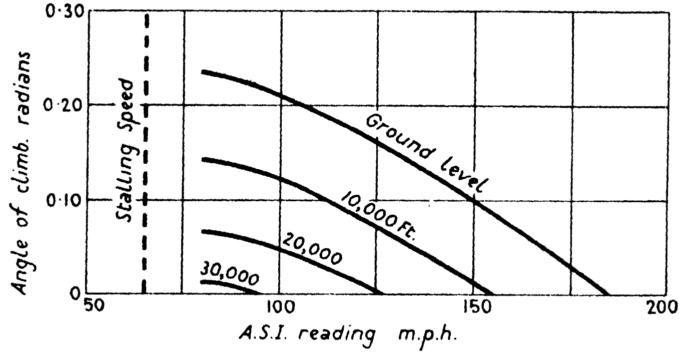


FIG. 12. The variation of the angle of climb with the A.S.I. reading.

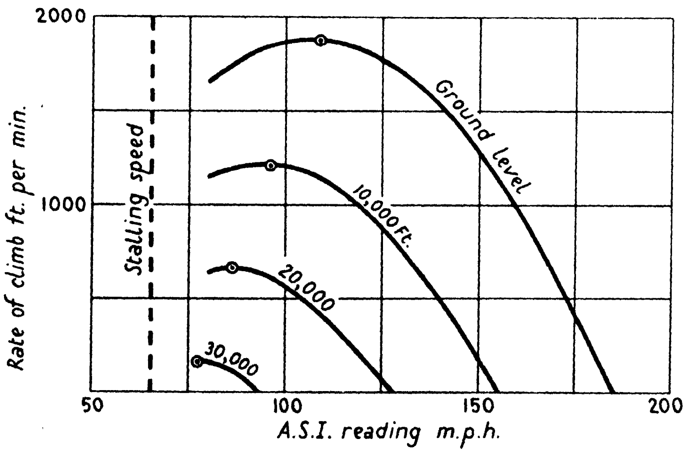


FIG. 13. The variation of rate of climb with the A.S.I. reading.

therefore 3,500 lb. The engine is taken to be unsupercharged, and its power to vary directly as the atmospheric density.

In figs. 12 and 13 the deduced angles and rates of climb are plotted against A.S.I. reading, and in figs. 14 and 15 against true speed. In fig. 16 the 'maximum performance' (speed and rate of climb) is shown against height.

The general nature of the full throttle performance curves shown in fig. 16 is typical of an aeroplane fitted with an unsupercharged engine. The top speed in level flight falls as the height increases, at

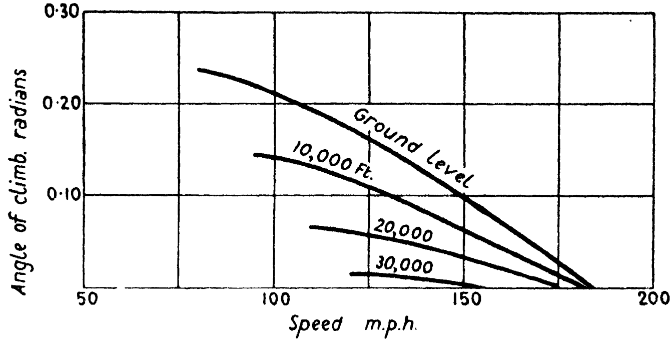


FIG. 14. The variation of the angle of climb with the true speed.

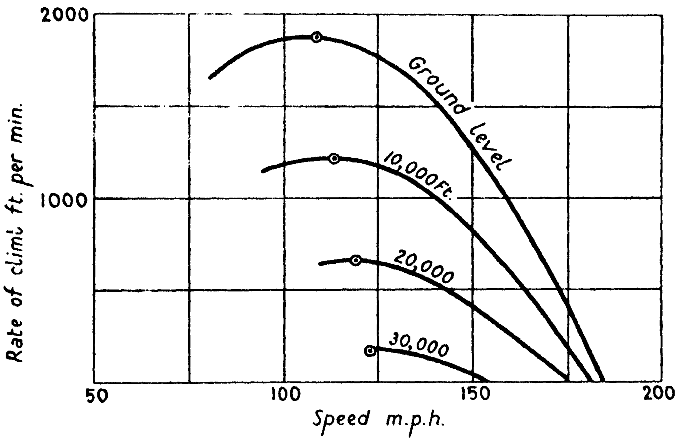


FIG. 15. The variation of the rate of climb with the true speed.

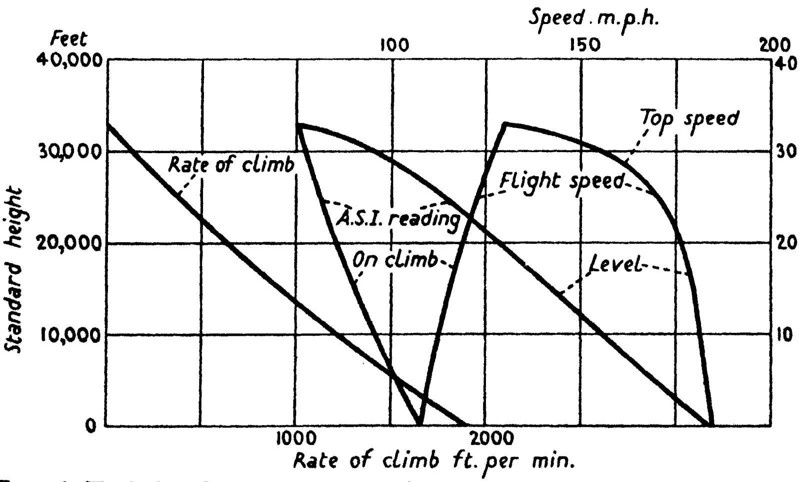


FIG. 16. Typical performance curves, maximum rate of climb, and forward speed at different heights.

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first slowly and then more rapidly. The fall in the maximum rate of climb is almost in proportion to the height. The maximum rate of climb is attained at an A.S.I. reading which decreases, although the corresponding flight speed increases, with height. The former is approximately proportional to $\sigma^{0{\cdot}33}$, the latter therefore to $1/\sigma^{0{\cdot}16}$. These relations are empirical, but are representative of average aeroplanes and engines.

In one respect fig. 16 gives too optimistic a picture. There is underlying it the assumption, introduced at the end of art. 11, that the engine power is directly proportional to the relative atmospheric density σ and to the rotational speed n, or, in other words, that the engine torque is proportional to σ and independent of n. In practice the torque of an unsupercharged engine, apart from small changes caused by variations of n, falls appreciably more rapidly than in proportion to the density of the surrounding atmosphere, for reasons explained in Chapter XII. Much time has been devoted to finding a general relation which would represent its variation sufficiently well for prediction of aeroplane performance, and while the results cannot be said to be conclusive—partly on account of the difficult nature of the experiments and the involved technique of their reduction, but also because in all probability no such general relation applicable to all engines exists—it may be assumed as a rather closer approximation that the power varies in proportion to the product of the density and the square root of the absolute temperature. In the standard atmosphere, in which $\theta \propto \sigma^{0{\cdot}235}$, this means that the engine power would be proportional to $\sigma^{1{\cdot}12}$. Thus at 10,000 ft. where $\sigma = 0{\cdot}738$ the power of an unsupercharged engine would be $0{\cdot}712$, and at 20,000 ft. where $\sigma = 0{\cdot}533$ it would be $0{\cdot}495$, of its ground level value. The calculations made above and summarized in fig. 16 therefore over-estimate the power available at these heights by 4 and 7 per cent. respectively, and the actual performance would therefore be appreciably less than that given by fig. 16.

There is no special difficulty in following through the calculations *ab initio* using the more representative law of power variation, but since the relation between V and n (fig. 7) will not now be the same at all heights the labour is considerably increased. This may be avoided by making use of the following general result, which has also the merit that it enables the consequences of *any* law of power variation to be determined in general terms, irrespective of the characteristics of any particular aeroplane or engine.

The atmospheric density σ and temperature θ are fixed by convention in the 'standard atmosphere', in which they are therefore functions of the height h. If the engine power varies in any systematic

way with density and temperature (say as $\sigma^a\theta^b$) the ratio e in which it is reduced at any height h will also be expressible, in a standard atmosphere, as a function of σ, namely as $\sigma^{(a+0\cdot235b)}$, and also, if desired, as a function of the height h. It can be shown that if performance calculations for the height h are made as though the density there were not σ but e (but otherwise exactly as above, i.e. taking the power as varying in proportion to the fictitious 'density' e), then the *angle* of climb at any A.S.I. reading will be given correctly, but in order to calculate the rate of climb and flight speed the correct value, σ, of the density must be used. This is equivalent to the following working rule:

Calculate the complete performance on the assumption that the engine power is proportional to σ, giving a top speed in level flight V and a maximum rate of climb v_c, at a height h_1, where the standard density is σ_1. If the engine power law is $e = \sigma^{1+m}$, then the top speed and maximum rate of climb at a height h_2 where the density is $\sigma_2 = \sigma_1{}^{1/(1+m)}$ are found by multiplying V and v_c by the factor

$$\sqrt{\frac{\sigma_2}{\sigma_1}} = \sigma_1^{m/\{2(m+1)\}} = \sigma_2^{\frac{1}{2}m}.$$

The A.S.I. reading, both at top speed and at maximum rate of climb, and the value of V/n, will be unaffected.

It will be observed that the rule cannot be framed in the form of a percentage reduction of performance *at a given height*. But by inserting the conventional relation between height and density the following more convenient empirical generalizations are found to hold:

Calculate the top speed and rate of climb at a height h on the assumption that the engine power is proportional to σ; if the power varies as $\sigma^{1\cdot1}$ reduce the height by 7·5 per cent. and the performance by $1\cdot5h$ per cent. (h being measured in units of 10,000 ft.): if the power varies as $\sigma^{1\cdot2}$ reduce the height by 15 per cent. and the performance by $2\cdot5h$ per cent.

The performance curves of fig. 16 modified in this way are given in fig. 17. In practice the actual performance would lie between the curves corresponding to $e = \sigma^{1\cdot1}$ and $e = \sigma^{1\cdot2}$. A further correction may be made along the same lines if the power is not proportional to n (i.e. if the torque is not constant) but this effect is in practice small and outside our present scope.

The effect of supercharging on performance may be deduced without difficulty. At any fixed throttle setting, an engine fitted with a centrifugal blower in the induction system, running at a constant multiple of the crankshaft speed, is not in any essential

respect different from an unsupercharged engine. Its torque will be approximately independent of n and proportional to σ^{1+m} where m is between 0·1 and 0·2. But below a certain 'rated height' the absolute value of the power produced will exceed that which the cooling system can deal with, and the engine must therefore be throttled progressively, the pressure in the induction system (the so-called 'boost' pressure) not being allowed to exceed that at the 'rated height'. This has the result that the maximum torque is

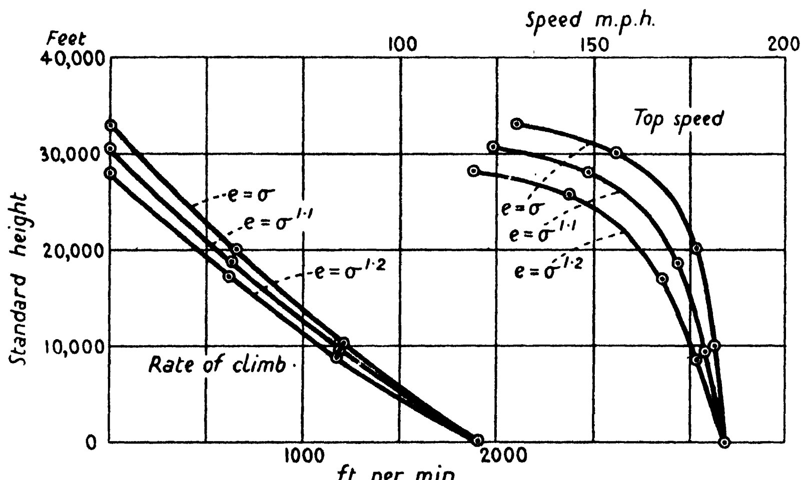

FIG. 17. Performance curves with different assumptions as to the power factor of the engine.

nearly constant at all heights below the rated height. If the aeroplane of fig. 16 is fitted with an engine supercharged to a rated height of 10,000 ft. and giving the performance of fig. 16 at 10,000 ft., then above that height it will have the same performance as with the unsupercharged engine, but below it will have a performance corresponding roughly to a torque independent of height (see art. 77 for the true variation of torque). It follows from certain relations to be given in the next article, and in particular from II (3), that the A.S.I. reading at top speed will be constant at all heights below the rated height, so that the corresponding flight speed will *fall* in proportion to $\sigma^{-\frac{1}{2}}$. It can readily be shown that the same conclusion applies to the rate of climb below the rated height. In fig. 18 is shown the performance corresponding to two assumed supercharged engines, one giving the same power as the unsupercharged engine of figs. 16 and 17 at 10,000 and the other the same power at 20,000 ft. respectively. The piston displacements of these alternative engines

would be approximately 75 and 55 per cent. of the unsupercharged engine. The 'potential ground-level power' would be the same for all three, but could not be realized with the smaller, supercharged engines, owing to the impossibility of cooling the cylinders.

The above generalizations concerning supercharging do not apply exactly in practice, since for reasons which will be dealt with in Chapter XII the maximum torque below the rated height is not exactly constant. But no serious error is introduced by disregarding this in a short survey of the effect of supercharging on performance.

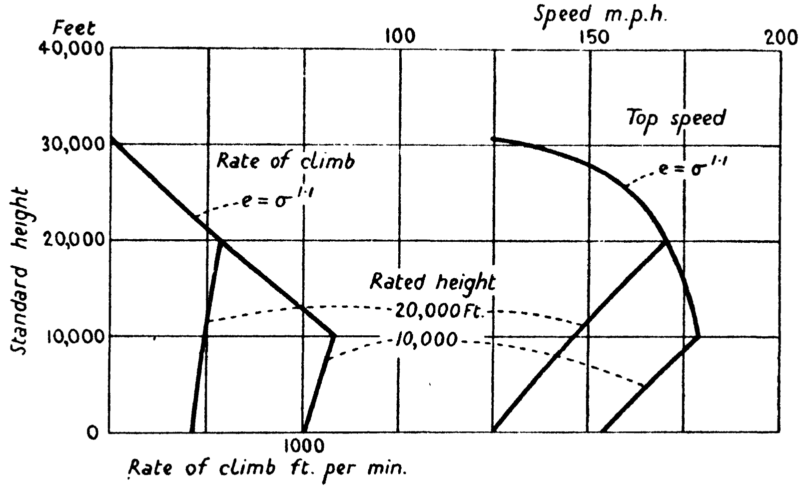

FIG. 18. Performance curves for an aeroplane with a supercharged engine.

ART. 13. *Characteristics peculiar to level flight—Range and altitude.*

One feature of the 'performance' of particular importance from the point of view of the engine remains. The results of figs. 10 to 18 refer to full-throttle conditions, and give therefore the maximum, but by no means the most economical, performance, especially in level flight. It is naturally in relation to the power required for level flight that the specific fuel consumption of the engine is of most interest.

In level flight, equations (4) and (5) reduce to $T = D$ and $P = H$ respectively. We have already considered a special case, namely, level flight at top speed, i.e. with full throttle, and it follows from figs. 5 and 6 that in level flight at lower speeds the throttle opening must be reduced progressively. As mentioned in art. 12, variation of the throttle opening is equivalent to substituting a different engine, and would therefore appear to involve repeating the analysis represented by fig. 7 at each speed. We can avoid this, however,

and deal with the problem by a general argument, from which it will be seen that deductions of great interest concerning economy in flight at various heights can readily be made.

The expression of the fundamental properties of an airscrew by the two relations shown in figs 7A and 8A, can be amplified by stating the constancy of a series of factors under certain conditions:

(I) *At any given value of V/n the following characteristics have fixed values:*

(1) $\dfrac{Q}{\rho n^2}$,

(2) $\dfrac{Q}{\rho V^2}$, since it equals $\dfrac{Q}{\rho n^2} \times \left(\dfrac{n}{V}\right)^2$,

(3) η,

(4) $\dfrac{T}{\rho n^2}$, since it equals $\dfrac{Q}{\rho n^2} \times \eta \times \dfrac{2\pi n}{V}$,

(5) $\dfrac{T}{\rho V^2}$, " " " $\dfrac{T}{\rho n^2} \times \left(\dfrac{n}{V}\right)^2$.

Now since in level flight $T = D$, and since D depends only on ρV^2 (eqn. (9)), it follows that T (and hence $T/\rho V^2$) is fixed at each value of ρV^2 (or V_i^2); and hence that at each value of the A.S.I. reading, V_i, *whatever the value of* ρ, i.e. *whatever the height*, there is in level flight one value and one value only of the advance per revolution V/n. From this, by use of the above relations I (1) to (5), we deduce that

(II) *In level flight at any given value of the A.S.I. reading the following quantities are fixed, irrespective of altitude,*

(1) the drag of the aeroplane, D,
(2) the thrust of the airscrew, T,
(3) the torque of the airscrew, Q,
(4) the advance per revolution, V/n,
(5) the efficiency of the airscrew, η.

Since, however, the actual speed V depends on the air density, being in fact (eqn. (3)) equal to $V_i/\sigma^{\frac{1}{2}}$, it follows from II (4) that the rotational speed n rises, as the air density falls, in proportion to $1/\sigma^{\frac{1}{2}}$.

These relations have a profound influence on the working of an aero-engine in level flight and will be referred to later (Chs. XII and XIII). For our present purpose we proceed as follows: from I (5) we can plot fig. 19A, representing $T/\rho V^2$ against V/n, and from equation (9) or fig. 5 we can plot fig. 19B representing $D/\rho V^2$

against V_i. By the construction shown in broken lines (representing $T/\rho V^2 = D/\rho V^2$) we obtain the unique relation between V/n and V_i shown in fig. 20 and referred to in II (4) above, and from the known relation between V/n and η (fig. 8 A, representing I (3)) we deduce

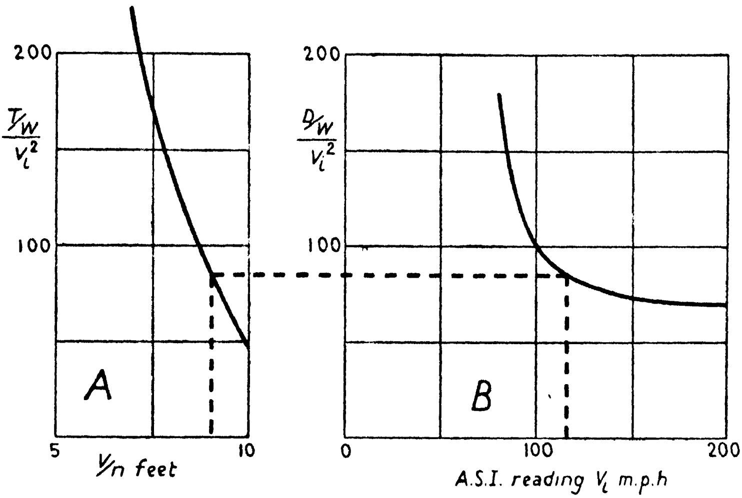

FIG. 19. The relationship between V/n and the A.S.I. reading in level flight.

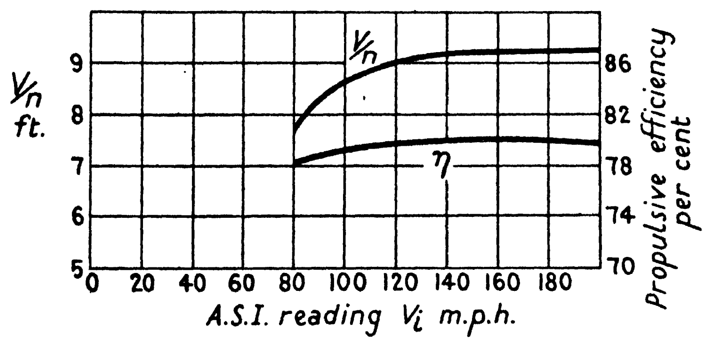

FIG. 20. The variation of airscrew efficiency and V/n with A.S.I. reading in level flight.

the unique relation between η and V_i which exists in level flight (see II (5)) and is shown by the lowe. curve in fig. 20.

The most striking feature of these results is the small range over which V/n and the efficiency vary in level flight. For example, with the above characteristics, η lies between 79 and 80 per cent. from 100 to 200 m.p.h. and falls only to 78 per cent. at 80 m.p.h. Consideration of fig. 19 B will show that η would be constant if $D/\rho V^2$ were constant. This would be true if the induced drag were zero. In fact, if the drag of an aeroplane were strictly proportional to ρV^2

then in level flight the efficiency of the screw would be constant and the rotational speed would be proportional to the flight speed.*

The foregoing results lead directly to important conclusions concerning the rate of consumption of fuel and hence the range of flight of an aeroplane. If the specific fuel consumption of the engine is p lb. per B.H.P. hour when it is developing the power Z required for level flight at any specified height (defined by σ) and flight speed V m.p.h., the fuel consumed per mile covered (relative to still air) is pZ/V lb., which can readily be shown to be equal to $\frac{1}{375}pD/\eta$. Now from the set of relationships given under (II) above it follows that D/η depends only on the A.S.I. reading. In table 8 its values over the whole range of A.S.I. readings are given, divided by the total weight W in units of 1,000 lb., together with other relevant quantities. The values of η are those of the 'propulsive' efficiency. From this table may be deduced the whole of the effect of the characteristics of the aeroplane and airscrew upon the rate of consumption of fuel.†

TABLE 8

Value of characteristics in level flight at all heights.

(1) A.S.I. reading $V_i = V\sqrt{\sigma}$	(2) Drag D/W lb. per 1,000 lb.	(3) Advance per revolution V/n ft.	(4) Efficiency η	(5) $\frac{D}{W\eta}$	(6) Power $\frac{Z\sqrt{\sigma}}{W}$	(7) Rotational Speed $n\sqrt{\sigma}$ r.p.m.	(8) Torque
80	95	7·7	0·78	122	26·1	910	0·365
100	100	8·6	0·79	127	33·9	1,020	0·400
120	119	9·0	0·795	150	48·0	1,120	0·495
140	148	9·2	0·80	185	69·0	1,340	0·623
160	184	9·25	0·80	230	98·0	1,520	0·779
180	227	9·28	0·80	284	136	1,710	0·960
200	276	9·30	0·80	345	184	1,890	1·14

Notes. (1) From cols. (1), (6), and (7) the flight speed, power per 1,000 lb., and rotational speed at any height are found by dividing by $\sqrt{\sigma}$.

(2) Cols. (2), (3), (4), (5), and (8) apply at the A.S.I. readings of col. (1) at all heights.

(3) Cols. (3) and (7) refer to a screw diameter of 8·8 ft., suitable for an aeroplane of total weight 3,500 lb. For an aeroplane of the same resistance characteristics but of a different total weight the appropriate screw diameter, and V/n, will vary directly, and $n\sqrt{\sigma}$ inversely, as the square root of the weight.

(4) The unit of torque, col. (8), is the maximum torque available from the engine at ground-level.

The effect upon the specific fuel consumption p of the various conditions under which the engine may have to work when in flight

* This is nearly true of an airship.

† It is necessary to note that the aeroplane is assumed to have a fixed total weight. The effect of an appreciable change in weight due to consumption of large amounts of fuel can be dealt with along obvious lines.

will be dealt with fully in Chapter XIII, but for our present purpose it is sufficient to anticipate the main conclusion of that chapter by the generalization that while the range of values of p within which an engine will run is very wide, the *minimum* value attainable with safety varies but little from engine to engine, and hardly at all for a given engine between, say, 40 and 80 per cent. of the maximum (ground-level) torque, at all relevant rotational speeds and at all heights. We may therefore tentatively regard p as fixed and take a round figure of 0·5 lb. per B.H.P. hour as representative of an efficient aero-engine.

It has been shown that the fuel per mile is proportional to pD/η and that D/η is constant so long as the A.S.I. reading is constant; and hence we may at once draw the following conclusions:

(III) (1) the rate of fuel consumption of an aeroplane (lb. per mile) depends only on the A.S.I. reading V_i and, at a given A.S.I. reading, is independent of height;

and (2) the minimum rate of fuel consumption, which occurs at the A.S.I. reading for which D/η is least, is independent of height except in so far as p may vary with height;

but (3) the time taken to fly a given distance at any given rate of fuel consumption per mile decreases with height in proportion to $\sigma^{\frac{1}{2}}$, since the speed of flight V is equal to $V_i/\sigma^{\frac{1}{2}}$.

The example taken in the foregoing pages is typical in that it leads to an A.S.I. reading for maximum economy (85 m.p.h.) which is clearly too low to be of direct practical significance, having regard to the average wind which must be allowed for in transport over long distances. At 10,000 ft., however, the corresponding flight speed is 99 m.p.h. and at 20,000 ft., 116 m.p.h., emphasizing the advantage of flying high. But these speeds are still low for the type of machine considered, and it is instructive to consider the problem on the basis of a fixed (true) speed of flight of, say, 150 m.p.h. at all heights. The results, which may be readily deduced from table 8, are shown graphically in fig. 21, which includes also the data relating to flight at top speed.* Fig. 21A shows the A.S.I. reading and flight speed corresponding to (1) the top speed, (2) 150 m.p.h., and (3) the most economical speed; fig. 21B the corresponding engine speed; fig. 21C the rate of fuel consumption per mile as a percentage of the minimum rate, and the corresponding percentage throttle opening, i.e.

$$(\text{torque required} \times 100) \div (\text{maximum available torque});$$

* At top speed (full throttle) the specific consumption will be considerably in excess of the minimum value assumed above. The corresponding consumptions on fig. 21C are therefore optimistic to the extent of at least 25 per cent.

and fig. 21D the torque itself.* It will be seen that at 150 m.p.h. the consumption is at ground-level 70 per cent. above the minimum, whereas at 10,000 ft. the excess is reduced to 35 per cent., and at

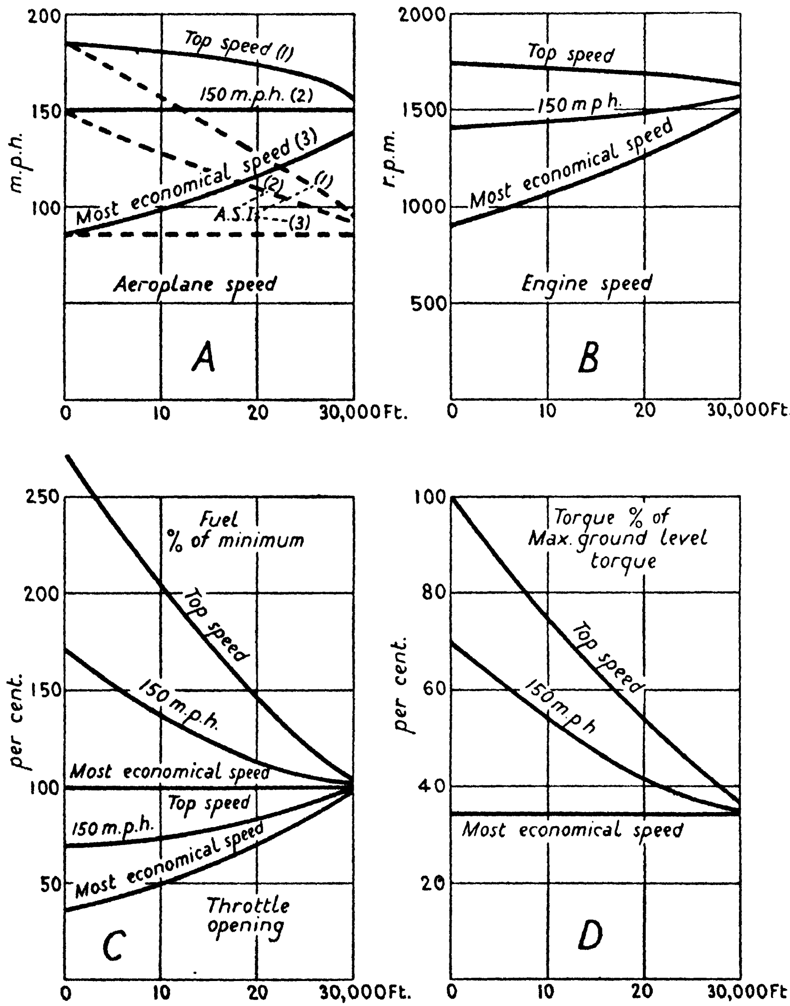

Fig. 21. Fuel consumption per mile in level flight at different heights and speeds.

20,000 ft. to 12 per cent. There is clearly a great advantage in flying high when the speed of flight is fixed.

So important is the time factor in the problem that it is useful to generalize the above results, with some slight sacrifice in accuracy, as follows. From table 8 it will be seen, as has already been empha-

* As a percentage of the maximum torque available from the engine at ground-level.

sized, that the average airscrew efficiency varies so little in level flight that its effect may be disregarded. Fuel consumption per mile then becomes identified with resistance, and maximum economy with minimum resistance. Now the resistance may be expressed (eqn. (9)) in the form $A/V_i^2+BV_i^2$ and it has already been remarked that the total is least when the two parts are equal, namely when

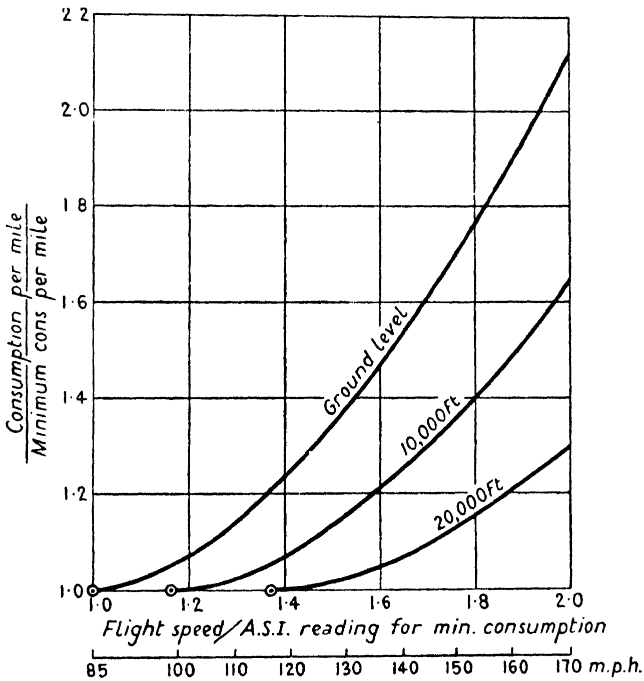

FIG. 22. Ratios of fuel consumption per mile to the minimum consumption at different heights and speeds.

$V_i^4 = A/B$. It may readily be shown that the drag at a speed $(1+x)$ times that at which the drag is least is $\left[1+\frac{1}{2}\left\{\frac{x(x+2)}{x+1}\right\}^2\right]$ times the minimum drag, and the consumption per mile bears the same ratio to the minimum consumption. The speeds referred to are A.S.I. readings. At any height corresponding to a relative density σ the consumption will be the same at a flight speed $1/\sigma^{\frac{1}{2}}$ times the A.S.I. reading. These relations are summarized in fig. 22, the abscissa being the ratio of the true flight speed to the (constant) A.S.I. reading characteristic of maximum economy. A scale of m.p.h. applicable to the particular case dealt with above is attached. The points marked in the circles are the speeds for maximum

economy at the three heights chosen. The whole range of speed covered by the curves is within practicable limits for the machine considered, its top speed lying well above 170 m.p.h. up to 20,000 ft. From this diagram, which (except for the scale of m.p.h.) may be taken to apply to all aeroplanes, the effect of speed and height on fuel consumption may be estimated without serious error. As a guide to the scale of abscissae and ordinates, the unit of fuel consumption, per lb. per 100 miles per 1,000 lb. of total weight, is approximately $0{\cdot}33 \times \sqrt{AB}$ (taking $p = 0{\cdot}5$ lb. per B.H.P. hour and $\eta = 80$ per cent.) where A and B are as defined above, namely the induced and (profile+parasite) drag per 1,000 lb. at 100 m.p.h., while the unit of speed is $100\,(A/B)^{\frac{1}{4}}$ m.p.h. Thus in the example taken $A = 33$ and $B = 67$ (see eqn. (9)), giving a minimum fuel consumption of $0{\cdot}33\sqrt{(32 \times 67)} = 15{\cdot}6$ lb. per 100 miles per 1,000 lb. at a speed of $100(\frac{33}{67})^{\frac{1}{4}} = 85$ m.p.h. The corresponding ton-miles per gallon (at 7·8 lb. per gallon) is found by dividing the former figure into 350, viz. $\frac{350}{15{\cdot}6} = 22{\cdot}5$ ton-miles per gallon.

It is once again apparent that the *essential requirement for good economy at high speed is low parasite drag* which corresponds to a low value of B. For example, if the aeroplane considered had its parasite drag reduced by 50 per cent., reducing B from $17+50$ to $17+25 = 42$, the above figures would become 12·4 (28 ton-miles per gallon) and 94 m.p.h.

III

THE LIMITS OF INTERNAL COMBUSTION ENGINE PERFORMANCE

ART. 14. *On cylinder size, and the factors which govern power output.*

In the foregoing chapter the aero-engine has been regarded merely as a machine which is capable of providing a certain torque Q_E lb. ft. sufficient to drive an airscrew at a speed of revolution n.

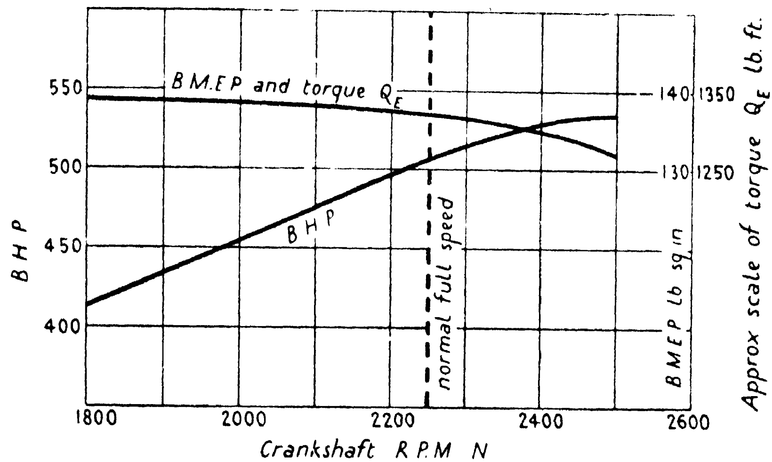

FIG. 23. Curves of power, B.M.E.P., and torque at the crankshaft, in lb. ft., for Rolls-Royce Kestrel engine. 12-cylinder V-type. C.R. = 6. Normally aspirated. Bore and stroke = 5 in. × 5½ in.

If the engine has a reduction gear between its crankshaft and the airscrew of ratio n/N where N is the crankshaft speed, then

$$Q_E = N/n \text{ (the mean torque at the crankshaft).}$$

If the throttle position, or the air density in which the engine is working, be altered, then the engine gives a new relation between Q_E and N; but with fixed engine controls (throttle, ignition timing, and fuel-air mixture) and constant air density, Q_E is a definite function of N which depends only on the engine design. This relation between Q_E and N cannot be expressed exactly by any simple formula, but a typical example is shown by the upper curve in fig. 23. Over a limited range of speed Q_E would not vary widely, and for some purposes it is sufficiently near the truth to regard it as being constant.

The rest of this book will be largely concerned with the variations of Q_E, or more usually with the brake mean effective pressure

(B.M.E.P.) which is proportional to Q_E, under all practicable circumstances; and the present chapter will be devoted to examining in what ways the B.M.E.P. of a given cylinder working at ground-level can be made as great as possible, and what are the factors that limit its further increase.

The B.H.P. of an engine is proportional to the product of the speed, the total swept volume of the cylinders, and the B.M.E.P.; and all three factors are to some extent interdependent. The B.M.E.P. will vary with the speed, as illustrated in fig. 23, owing to changes of volumetric efficiency; and the maximum B.M.E.P. which can be maintained in any engine without overheating, and hence the power per unit of swept volume, will in general be greatest in small cylinders. The lightest type of engine might therefore be expected to be that in which a maximum number of pistons operate on a minimum number of cranks: an ideal which is well met in the radial engine where as many as nine can be made to operate on one crank.

Before discussing methods of increasing the B.M.E.P. in any given cylinder it will be advisable to introduce a digression upon the question of cylinder size, and why it affects the limit of power obtainable per unit of swept volume. And to point out, moreover, what is often overlooked, that a figure for power output expressed as h.p. per litre* is totally misleading as a comparative figure of merit if applied to cylinders of different sizes. In two cylinders of equal merit, geometrically similar in design but of different sizes, the h.p. per litre will be inversely proportional to their linear dimensions.†

Apart from this purely dimensional reason, however, why small cylinders give high values of the h.p. per litre, they have certain advantages associated with the question of heat-flow. The problem of getting rid of the waste heat from every cu. ft. of the cylinder gases becomes very rapidly more difficult as the cylinder size is increased. A small combustion space and a high rotational speed are both advantageous from the point of view of avoiding detonation, and very high M.E.P.s can, therefore, more easily be maintained in a small cylinder without introducing this prime cause of overheating.

In a practical design of aero-engine, in which a certain aggregate horse-power is required, combined with a minimum weight and bulk per h.p., a balance has to be struck between the desirable concentration of power made possible by having many small cylinders, and the drawback of excessive complication and cost of manufacture

* It would be more consistent to express power output as 'B.H.P. per 100 cu. in.' (100 cu. in = 1·64 litres), but the litre has been fairly generally accepted as the unit in this connexion, and it seems preferable to follow custom in the matter.

† For proof of this, see Appendix I, p. 387.

involved in a large number of cylinders each complete with its valve-gear and a multiplicity of moving parts.

The result of this compromise has been that aero-engines, if we exclude the 100–150 B.H.P. class as well as special designs for racing purposes, range from those with 16 cylinders of 3½ in. × 3½ in., bore and stroke, and a normal speed of 3,500 r.p.m., to an upper limit of cylinder size about 6 in. × 7½ in., and 1,800 r.p.m., with a corresponding range of power output from 350 to about 900 B.H.P. It is exceptional for an engine of 400 h.p., or over, to be designed with a cylinder bore of less than 5 in.

In the small engine mentioned above the output per litre of swept volume works out at 34 B.H.P., as against about 18–19 B.H.P. in the large engines; but any attempt to obtain 800–900 B.H.P. from an engine with cylinders of a size 3½ in. × 3½ in., would lead to an almost prohibitive mechanical complexity.

The above figures refer to the normal ranges of aero-engines. It is of interest in this connexion to quote the extreme figures obtained upon the Rolls-Royce engines built for the Schneider Trophy race in 1931. In those the estimated B.H.P. per litre was no less than 64, obtained by a combination of high speed, supercharging, and the use of a special fuel.

The cylinder size in those engines was 6 in. × 6·6 in., and even their astonishing figure has been far surpassed by a very small cylinder, 2⅞ in. × 3½ in., of a purely experimental type designed by Ricardo. When heavily supercharged and run at speeds ranging from 5,000 to 6,000 r.p.m., this single-cylinder engine achieved an output of no less than 107 B.H.P. per litre. It was a sleeve-valve cylinder, so that in spite of a high degree of supercharge there was no red-hot exhaust valve to bring on detonation. This fact, and the small size of the piston, made it possible ~~to employ~~ the unusually high compression ratio of 6·8 to 1, and so to maintain it as a highly efficient cylinder in which the proportion of waste heat was small. The amount of heat passing through the cylinder per minute was phenomenal, but the temperature of the exhaust gas was not excessive, and the engine proved itself capable of running steadily for 100 hours or more at B.M.E.P.s of the order of 200–250 lb. per sq. in.

To achieve a high M.E.P. two things are necessary: a high thermal efficiency and a large amount of heat generated per unit of cylinder volume. The heat generated will be proportional to the weight of the fuel-air mixture, so that the second requirement is equivalent to the need for a high charge-weight per cycle. The charge-weight may be increased by (*a*) lowering the temperature, and (*b*) increasing the pressure, of the fuel-air mixture supplied;

while the efficiency is simply a question of the expansion ratio employed—provided always that the fuel-air ratio and ignition-timing are adjusted for maximum power.

The use of a supercharger, supplying air and fuel at a high pressure, is the usual way of increasing the charge-weight per cycle, which we shall consider in detail later; but before doing so it will be useful to examine the possibilities of an increase of the B.M.E.P. in a normally aspirated cylinder by increasing the volumetric efficiency without any increase of the charge pressure, that is to say by a lowering of the charge temperature.

In a normally aspirated cylinder, supplied with an unvarying mixture of fuel and air, the indicated power output at a given speed depends only upon the thermal and volumetric efficiencies, and if we exclude pressure supercharging we are thrown back upon a lowering of the mixture temperature as the sole means of increasing the volumetric efficiency.* The thermal efficiency will depend upon a proper adjustment of the ignition timing and other details, but if we assume the adjustment for optimum conditions to be made, then the only way of increasing the thermal efficiency is by an increase of the expansion ratio.

In the following two articles the range of possible increases of the M.E.P. in a given cylinder at constant speed, through changes of compression ratio and of mixture temperature, will be reviewed.

ART. 15. *Compression ratio and power output.*

The fact may be recalled here that a change of compression ratio may itself alter the volumetric efficiency of a cylinder (see art. 55 (i)),† but in high-speed engines with valve overlap the effect is probably very small; and in any case this would not affect the soundness of examining the effect of one variable at a time in its influence on power output. In what follows, therefore, it will be assumed that the same mass of fuel-air mixture and of potential heat energy is drawn into the cylinder per cycle in all circumstances. When this is so the only way of increasing the power output is to raise the expansion, and therefore the compression ratio.

The advantage of raising the compression ratio, as a method of increasing the power output, is that the additional power is pure

* The ratio $\dfrac{\text{mass of mixture supplied per cycle}}{\text{mass to fill the swept volume at N.T.P.}}$ is the important figure for supercharged, just as for normally aspirated, engines, and the fact that for a supercharged engine the ratio will have a value greater than unity need not preclude the use of the term volumetric efficiency. Pressure supercharging may therefore be regarded as merely another way of increasing the volumetric efficiency of a cylinder.

† References in this form refer to vol. i of this book.

gain, derived from an improved thermal efficiency and without any increase of the fuel consumed; and furthermore that with the higher thermal efficiency there is less waste heat to be got rid of, and less danger of over-heating at a given power output.

With volatile liquid fuels we are limited in respect to the compression ratio to about 8 or 9 to 1 by the danger of pre-ignition; and we are further limited by detonation, according to the fuel available, to anything between 4 : 1 and 7 : 1.

Referring to fig. 60 (i) it will be seen that the I.M.E.P. of the Ricardo E. 35 single-cylinder engine at 1,500 r.p.m. and C.R. = 5 was 136 lb. per sq. in. when operating on the maximum power mixture, 20 per cent. rich. In those tests, heat to the amount of 0·024 C.H.U. per rev. was added electrically to the in-going air; this being the standard figure adopted by Ricardo as representing on an average the heat received from the induction system in a multi-cylinder engine. We may accept the figure of 136 lb. per sq. in. as typical for an engine of C.R. 5 : 1, while noting from fig. 60 (i) that it corresponded to a volumetric efficiency of 74·5 per cent.,* and that in any other engine of the same compression ratio, and with properly adjusted mixture strength and ignition timing, the I.M.E.P. would be higher or lower in proportion to the volumetric efficiency.

At higher or lower compression ratios, the I.M.E.P. will be in proportion to the indicated thermal efficiency, and from the known variation of efficiency with compression ratio, as given in fig. 39 (i), we may calculate the I.M.E.P. and hence the percentage increase or decrease of power output from a cylinder to be effected by a change of compression ratio alone. The figures are given in table 9, and

TABLE 9

I.M.E.P. obtainable at different compression ratios, provided a non-detonating fuel is available. The assumed I.M.E.P. of 136 *lb. per sq. in. at C.R. =* 5 *corresponds to a volumetric efficiency of* 74·5 *per cent. (see footnote) and to the maximum power mixture strength,* 20 *per cent. rich.*

Compression ratio	*I.M.E.P. lb. per sq. in.*	*Ratio to I.M.E.P. at C.R. = 5*
4·0	122	0·90
5·0	136	1·00
6·0	147	1·08
7·0	157	1·155
8·0	166	1·22

* Volumetric efficiency is given here as that when referred to N.T.P. conditions (see art. 55 (i)). Such a figure of 74·5 per cent. would mean 78·5 per cent. if referred to external air conditions of normal pressure and 15° C.

illustrated by the curve *A* in fig. 24. It will be seen that the full scope for power increase in this way, taking 7 : 1 as the practicable limit of compression ratio, is only about 15 per cent. over the power at C.R. = 5.

These are laboratory results, deduced from tests on a single-cylinder engine at 1,500 r.p.m., and it may be asked how they compare with the results achieved in practice by high speed engines.

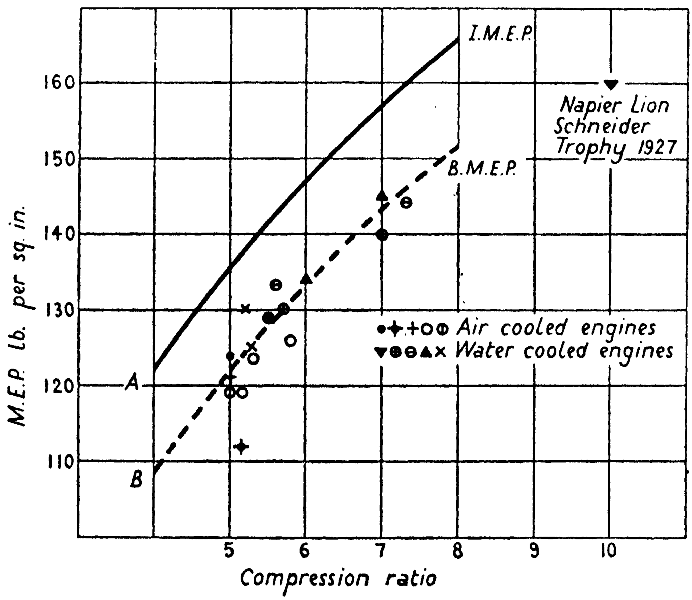

FIG. 24. Curves *A* and *B* show variation of I.M.E.P. and B.M.E.P. with compression ratio, based upon single cylinder experiments. Volumetric efficiency (referred to N.T.P.) 74·5 per cent. Mechanical efficiency assumed as ·90 per cent. Points are calculated from the rated B.H.P. of aero-engines.

Here one is met at once by a difficulty, in the first place because a figure for the I.H.P., the only proper basis for comparison, is seldom available, and still more because the effect of compression ratio on the B.H.P. is often masked by differences of volumetric and mechanical efficiency.

The normal speeds of engines differ widely, according to type and cylinder size, and among the various classes of aero-engines differences of speed have an influence upon the B.M.E.P.s achieved in practice which is often sufficient to obscure the effect of compression ratio.

Nevertheless it is of interest to record some of the B.M.E.P.s achieved by aero-engines in service, and this has been done on fig. 24.

It will be seen that the results are distributed fairly evenly about the dotted curve *B*, which represents the variation of B.M.E.P. of the single-cylinder engine for which the conditions at different compression ratios were properly comparable. The rise of B.M.E.P. with compression ratio is clearly of the same order, and variations above and below the curve may be put down to differences of speed, valve area and valve lift, stroke-bore ratio, and valve timing, all of which will affect the volumetric efficiency. The B.M.E.P.s will be influenced also, but to a less extent, by variations of the mechanical friction loss.

Wherever the observations are roughly comparable this has been indicated by the type of dot on the figure. Similar dots represent observations on the same type of engine and at nearly the same speed. It will be noticed that the higher ratios of compression are confined to water-cooled engines.

ART. 16. *Charge temperature and power output.*

We have seen in the last article that, with the limitation imposed by the quality of the fuels available, the possible gain of power from the use of a higher compression ratio is limited to about 15 per cent. above that at 5 : 1. In the present article we shall examine the possibilities of some further increase of power by a lowering of the charge temperature. In a later chapter the more general question of the influence of air temperature upon the power output of an aero-engine will have to be considered, and for the present we shall consider only an artificial lowering of the charge temperature as a possible means of increasing the power output from a cylinder of a given size.

Any lowering of the temperature of the mixture in the cylinder, at the moment when the inlet valve closes, means an increased density of the fresh charge and proportionately more heat generated in the cylinder per cycle. At a given compression ratio, therefore, the indicated power will be increased nearly in inverse proportion to the mean absolute temperature of the charge at the moment when the valve closes. And owing to the lowering of the mean gas temperatures during the cycle this extra power will be accompanied by some reduction of the danger from over-heating.

The only practicable way of lowering the charge temperature is by making use of the latent heat of evaporation of the fuel carried in with the air. In fig. 64 (i) there are given comparative curves for an engine of C.R. 5 : 1 when running with petrol and with alcohol as fuels. The curves show maximum I.M.E.P.s of 132 and 138 lb. per sq. in. respectively. These figures correspond to a difference in volumetric efficiency from 74 per cent. with petrol to 77·5 per cent. with

an alcohol mixture 30 per cent. rich, as illustrated in fig. 59 (i). This increase of volumetric efficiency is caused entirely by the large cooling effect due to evaporation of the alcohol fuel.

A volumetric efficiency of 77·5 per cent. does not represent all that can be done; for that was obtained with a certain amount of heat added to the carburettor, 0·024 C.H.U. per rev.; and moreover the fuel was ethyl alcohol containing only 1·5 per cent. of water. High as the latent heat of alcohol is, 226 C.H.U. per lb., that of water is 539 C.H.U. per lb., and some further addition of water, which combines very readily with alcohol, provides a fuel with a substantially greater cooling effect when it evaporates. By using a fuel consisting of alcohol plus 5 per cent. of water it was possible to obtain a maximum I.M.E.P. at C.R. 5 : 1 of 142 lb. per sq. in. with the same heat to the carburettor, and even 150 when no extra heat was added.

Alcohol, of course, will stand a much higher compression ratio than 5 : 1, and if full advantage be taken of this, as well as of its high latent heat, an I.M.E.P. of 170 lb. per sq. in. can be obtained in a normally aspirated engine.

The various possibilities are set out for comparative purposes in table 10, from which it will be seen that the use of an alcohol fuel

Table 10

I.M.E.P.s obtained upon the E. 35 engine at 1,500 r.p.m. with petrol and with alcohol, showing the effect of a high latent heat in increasing the volumetric efficiency and power output.

Fuel	*Compression ratio*	*I.M.E.P.* With added heat 0·024 *C.H.U./rev.*	*I.M.E.P.* With no added heat
Petrol	5	132*	140
Alcohol + 1·5 per cent. water.	5	138	146
Alcohol + 5 per cent. water .	5	142	150
Alcohol + 1·5 per cent. water.	7	156	165
Alcohol + 5 per cent. water .	7	161	170

with 5 per cent. of water is about half as effective in raising the M.E.P. as an increase of C.R. from 5 : 1 to 7 : 1. The latter gives a rise of about 20 lb. sq. in. and the fuel of high latent heat a further 10 lb. sq. in. An alcohol fuel being also a non-detonating fuel, a

* The engine in these tests was giving a rather lower I.M.E.P. than in those of table 9 (136 lb. per sq. in. under the same conditions) on account of an altered valve timing.

C.R. higher than 7 : 1 could no doubt be used with it, with an even higher M.E.P. than the best in table 10, but, apart from the high rate of consumption of such a fuel (0·6 lb. per B.H.P. hour even at C.R. = 7) and its high cost, it would hardly be possible to run a multi-cylinder engine upon alcohol without considerable heating of the induction system or, alternatively, some positive method of distributing the fuel equally between the cylinders. It was mentioned in art. 21 (i) that alcohol is exceptional among fuels in that, even with heat added, evaporation is liable not to be complete by the time the inlet valve has closed, and it would be almost impossible to achieve equal distribution of the fuel from a carburettor in the normal way when so large a quantity of it would have to pass into the cylinders as a liquid.

The use of alcohol by itself, therefore, in a multi-cylinder aero-engine as a means of getting a substantial increase of I.M.E.P. to, say, 170 lb. per sq. in., is no more practicable than it would be to aim at reaching such mean pressures, with more normal fuels, by employing extreme compression ratios of 8 : 1 or higher. As a method of increasing the charge weight per cycle, pressure supercharging is far more effective, and the important application of the use of fuels with a high latent heat is as an auxiliary to the pressure supercharger, to counteract some of the inevitable rise of temperature of the air on its way to the cylinders.

ART. 17. *Supercharging by an increase of charge pressure.*

In the last article the possibilities were examined of increasing the charge weight per cycle by a lowering of its temperature at constant pressure. In the present one we proceed to the alternative of increasing its pressure, not at constant temperature, but with as small a rise of temperature as conditions may allow. As already mentioned, the method of 'supercharging' by a lowering of the charge temperature would be preferable from nearly every point of view to the usual method, by increasing the pressure, if its possibilities were not so limited: engine temperatures would be lowered throughout; compression ratios could be raised, instead of having to be lowered; and, as a result, supercharging by lowering the charge temperature could be made to increase not only the power, but the thermal efficiency of the engine.

The possibilities along these lines, however, as set out in the last article, are so limited that even if the special fuels or other means of doing it were available, the method would still be of academic interest only compared with the possibilities of pressure supercharging. As against the 7 per cent. increase of power shown in table 10 when

using alcohol, over that with petrol at the same compression ratio, the pressure supercharger can supply air at 50 per cent. or even 100 per cent. above atmospheric pressure, with a more than proportional increase of power output. The two methods can, of course, be used in conjunction, and they were so used in the Rolls-Royce engines for the Schneider Trophy seaplane. In those engines air was supplied to the induction pipes at considerably more than two atmospheres pressure, and at the same time it was cooled by the use of a special fuel containing a large proportion of alcohol.

As distinct from supplying air at more than the ground-level atmospheric pressure, as in the Schneider Trophy engines, a supercharger may be employed only to counteract the fall-off of the atmospheric pressure with altitude. In this application of supercharging, for ground-power maintenance, the air pressure in the engine induction pipe is maintained constant at, or near, its normal ground-level value up to the 'supercharged height' of the aeroplane, which is the greatest height to which the supercharger is capable of maintaining that pressure when driven at its normal full speed.

While near the ground the supercharger is allowed to produce little or no increase over the external atmospheric pressure* and could with advantage be disconnected altogether. As the aeroplane rises the air-throttle is gradually opened, so that at constant r.p.m. an approximately constant weight of air would be delivered to the engine per minute; for the combined effect of the compression in the supercharger and the diminishing amount of throttling at the air-intake serves to keep the delivery pressure constant. The single-stage gear-driven superchargers in common use to-day are capable of maintaining ground-level pressure up to a height of about 12,000 ft., corresponding to a compression by the supercharger in the ratio of 1·43 : 1.

For the present, supercharging will be discussed in terms of an increase of the pressure of the air supply above normal atmospheric, commonly called 'ground-boosting'; and later, the more usual condition will be considered, when the surrounding atmospheric pressure is below the ground-level value. At reduced atmospheric pressures the natural rate of heat dissipation to the surrounding air is reduced in proportion to the atmospheric density; while if ground-level power is maintained, so also will the amount of waste heat be maintained which has to be dissipated. The result, therefore, is an upset of the normal balance at which the working temperatures are maintained. The cardinal fact to be remembered during a discussion of any type of supercharging is that an abnormal amount of heat is

* An exception is often made during the process of taking-off from the ground; but this will be dealt with later on.

going to be generated per cu. ft. of cylinder volume; and although the maximum gas temperature may be only slightly increased, the total amount of waste heat to be got rid of through the cylinder walls and exhaust system will be increased as compared with the means for dissipating it. All the problems of heat-flow will therefore become more acute, and if these are not to be prohibitive and to lead to overheating of the pistons and valves it is essential that the proportion of this waste heat to the total heat should be kept as low as possible: in other words that the thermal efficiency of the engine should be high.

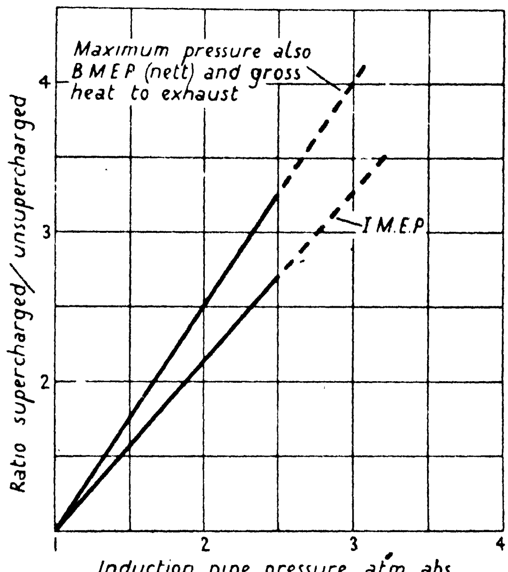

FIG. 25. Increase of maximum pressure, B.M.E.P., and gross heat to exhaust in a single-cylinder sleeve-valve engine when supercharged. The B.M.E.P.s have been corrected for the positive work done during the charging stroke.

It was emphasized in art. 61 (i) that a high thermal efficiency is more important from the point of view of minimizing the waste heat than because it means economy in fuel, and this is doubly so when, by supercharging, it is intended to increase the total quantity of heat which has to be dealt with per minute by an engine of a given size.

The only way of keeping a high thermal efficiency is to have a high expansion ratio, and one's first idea for a supercharged engine, therefore, would be to make it also a high compression engine. But against this there is the disadvantage of having high maximum pressures, and the necessity of employing a fuel with which detonation can be avoided. The maximum cylinder pressure in a normally aspirated engine of 6 : 1 compression ratio would be about 700 lb. per sq. in., and any substantial increase beyond that figure would mean a general stiffening up of the design, and enlargement of the bearing surfaces, undesirable in a light engine.

Fig. 25 shows how the maximum pressure, B.M.E.P., and gross heat passing away to the exhaust pipe all increased in a similar ratio, in a supercharged engine, and more rapidly than in proportion to the

induction pipe pressure. A threefold increase of the latter produced a fourfold increase of the three observed quantities, and the variation was linear throughout the range of pressures observed.

The I.M.E.P. also increased more rapidly than in proportion to the induction-pipe pressure, for reasons which will appear later, but less rapidly than the B.M.E.P. and the maximum pressure. The relation between the I.M.E.P. and B.M.E.P. reflects the rising mechanical efficiency of the engine as the power output was increased by the supercharging.

These results are based upon experiments by Ricardo with a single-cylinder engine, specially designed to withstand the huge gas forces produced by supplying air at up to 3½ atm. pressure; and owing to its very solid design the friction losses in the engine were abnormally large. The B.M.E.P. when normally aspirated was therefore unusually low, and it increased with supercharge rather more rapidly than it would in an engine of normal mechanical efficiency.

Later experiments with a poppet-valve cylinder more representative of aero-engine practice have shown that the rise of B.M.E.P., within a smaller range of induction pressures, was in that engine rather less than in proportion to the maximum pressure. The maximum pressure, B.M.E.P., and I.M.E.P. increased in the ratios 1·36, 1·32 and 1·29 when the induction pipe pressure was increased (at constant temperature) to 8 in. of mercury above atmospheric, i.e. in the ratio 1·27.

As mentioned already, the application of supercharging to the aero-engine is commonly restricted to achieving ground-power maintenance, and except in special engines for racing purposes no attempt is made to supply air to the cylinders at much more than ground-level pressure. In these circumstances the maximum cylinder pressures are kept down to about 800–900 lb. per sq. in. even when some supercharge at ground-level is allowed for a short time while taking off, and a reasonably high compression ratio of 5½ or 6 to 1 can be employed with the fuels generally available.

If an engine were designed to withstand continuous 'ground-boosting', it is clear from the results given above that those features which are related to the maximum cylinder pressures, such as crankshaft and connecting-rod bearings, would have to be enlarged roughly in proportion to the B.M.E.P., for the *same reliability*. An engine is designed with an ample margin of reliability under normal conditions, and its weight/h.p., therefore, can at any time be substantially reduced by ground-boosting; but only at a sacrifice of reliability. If this is to be maintained, the necessary enlargement of the bearing surfaces would react upon several major features of

design in such a way as seriously to minimize the benefit to the weight/h.p. ratio which might, at first sight, be anticipated. The general question of the value of ground-boosting as compared with the more restricted use of a supercharger for maintaining ground-level power can be better discussed after certain questions of heat-flow have been considered, for in practice these are usually more critical than questions of mechanical strength.

Even when the supercharger is employed only for ground-power maintenance the liability to detonation is greater in a supercharged than in a normally aspirated engine, owing to the higher temperature of the air supplied, after passing through the supercharger, even though the air pressure in the induction pipe may be the same in both. Detonation, therefore, limits the compression ratio which can be employed to a lower value than would be possible in a normally aspirated engine using the same fuel, and this all-important question must now be examined in some detail.

ART. 18. *Detonation in a supercharged engine.*

The correlation of the allowable supercharge pressure with the onset of detonation becomes very complex on account of the number of separate variables involved. The type of cylinder and sparking plug, the temperature of the mixture entering the cylinder, the ignition advance, and the type of fuel must all be specified in any quantitative examination of the possibilities of supercharging.

In practice the temperature of the mixture supplied would increase with the degree of supercharge, owing to the rise of temperature in the supercharger; and we have to decide, in any experimental investigation, whether to vary the temperature with the supply pressure or to vary each separately. It is known, also, that fuels with a high anti-knock value, if they owe this property to a large percentage of aromatic hydrocarbons, are adversely affected by high cylinder temperatures as compared with fuels which owe their anti-knock quality to the presence of naphthenes or tetra-ethyl lead. It follows that the effect of supercharge in promoting detonation will appear greater with a fuel rich in aromatics, than with the equivalent lead-doped fuel, for the process of pressure supercharging will necessarily produce a general increase of cycle temperatures. Then again, the investigation of supercharged operation is a research of great technical difficulty and some danger, where combustible mixture is to be supplied under a high pressure and measured at the same time.

No one has yet built an engine for supercharging research in which the compression ratio could be continuously varied, as in Ricardo's E. 35 engine; but perhaps the clearest way to envisage the

whole supercharging problem is to imagine such an engine as part of an experimental plant in which all the variables mentioned can be fixed or altered at will. Into the imagined working of this plant we can then fit the known experimental results.

Before reviewing these data, however, there is an important condition, peculiar to supercharging, which must first be mentioned. It was shown in art. 49 (i) that cooled exhaust gas artificially added with the in-going charge was very effective in suppressing detonation, and that when the normal proportion of residual gas was reduced by scavenging, violent detonation set in.

It was also pointed out that the ready suppression of detonation by throttling was due largely to the consequent increase in the proportion of residual gas to fresh charge, and that each 1 per cent. of residual gas would be responsible for a lowering of the temperature-rise on combustion by 20° C. or more.

In a supercharged engine, or an aero-engine in which ground power is maintained at altitude by a supercharger, the reverse of this state of things occurs; for the pressure of the residual gas left in the cylinder when the exhaust valve closes will be that of the surrounding atmosphere, while the fresh charge is at a higher pressure and density. The proportion of residual gas to fresh charge will therefore be substantially lowered below that in the normally aspirated engine at the same compression ratio.

Taking as an example two engines of 5 : 1 and 6 : 1 compression ratio in which the proportion by weight of residual exhaust gas when normally aspirated would be 7·4 and 6·0 per cent. respectively (see table 20 (i)), then the percentages when supercharged to various pressures will be as shown in column 3 of table 11.*

In arriving at these figures, and at those of column 4 of the table, several assumptions have to be made in regard to the heat picked up by the in-going charge on its way from the supercharger to the cylinder; but some error here will not affect the main conclusion to be drawn from table 11, that the drop in the percentage of residual gas as the supercharge pressure is raised will be a factor in promoting detonation of no less importance than changes of temperature and pressure before and during the compression stroke.

The figures show that with a supercharger supplying mixture at 1·5 atm. pressure to an engine of C.R. 5 : 1 the dilution has dropped from 7·4 to 5·1 per cent. This would allow an increase in the temperature-rise on combustion of about 50° C. The increase of temperature before compression due to the 0·5 atm. supercharge is seen from the table to be only 14° C.; this being because the rise of

* For the calculation of these figures, see Appendix II, p. 388.

TABLE 11

Proportions by weight of residual exhaust gas to fresh charge in a supercharged engine, and the mixture temperatures before compression for various degrees of supercharge.

Compression ratio	*Induction pipe press. atm.*	*Residual exhaust per cent. of fresh charge*	*Mixture temp. before compression*
5 : 1	1	7·4	122° C.
	1·25	6·0	127
	1·5	5·1	136
	1·75	4·5	144
	2·0	3·9	147
6 : 1	1	6·0	111° C.
	1·25	4·9	118
	1·5	4·15	128
	1·75	3·65	136
	2·0	3·2	141

temperature in the supercharger is to some extent compensated by the smaller proportion of hot exhaust gas which the fresh charge meets within the cylinder. An increase of temperature of 14° C. before compression would cause an increase of temperature of about 24° C. at the end; so that the increased *rise* of temperature in the supercharged engine, due to less dilution with residual gas, has much the more important influence on the maximum temperature of explosion.

If we make a similar comparison between the two compression ratios, 5 : 1 and 6 : 1, without any supercharge, the temperatures before compression can certainly be no greater at the higher ratio, and are probably some 10° C. less, as shown in table 11, so that even at the top of compression the higher ratio cannot be responsible for an increase of more than 20° or 30° C. The reduction of residual gas from 7·4 per cent. to 6 per cent. at the higher ratio would allow an increase in the rise of temperature on combustion of just about the same amount; so that, as between the two compression ratios, when unsupercharged, the increase of the initial temperature, and of the rise in temperature due to combustion, contribute about equally to the increase of final temperature.

In approaching, now, the results of supercharging experiments it will be well to state first what the aim of such a research would be. In the first place, with a given fuel known to be able to stand a certain C.R. in a normally aspirated cylinder, it is important to a designer to know by how much the C.R. would have to be lowered, to avoid detonation, if the same cylinder be supercharged to various pressures in the induction pipe. Secondly, he would need to know

if, and to what degree, detonation would be affected by the temperature of the mixture supplied by the supercharger.

The evidence on this last point is somewhat conflicting. In Ricardo's experiments, as we shall see, the conclusion is that an increase of temperature of the mixture, at the same supercharge pressure, does not increase detonation to an important extent; so that one is thrown back rather upon the increased rise of temperature on combustion caused by the altered dilution with residual gas, as the more important factor in promoting detonation in a supercharged cylinder. But Ricardo's experiments were made upon an engine with a sleeve valve, in which there would be nothing to correspond to the hot exhaust valve of the normal engine. The bulk of the evidence from poppet-valve engines would certainly be that a rise of temperature of the in-going mixture promotes detonation: whether directly, or through the effect which it has upon valve and piston temperatures, it would be difficult to say. A comparison of the results obtained with heated and unheated air in poppet-valve cylinders is to be given presently, in connexion with fig. 27.

Owing to the difficulties inherent in experiments upon supercharging it is usually not possible to observe directly the kind of data required by the designer, and therefore keeping in mind these requirements as set out above, the relevant experimental results must be critically examined and interpreted so as to yield the required information.

Ricardo[2] has given the results of a series of experiments upon his sleeve-valve engine of compression ratio 4·3 : 1, built strong enough to withstand supercharging to an induction pressure of 4 atmospheres. Since the compression ratio was fixed, the comparative ratios allowable when using the same fuel at different degrees of supercharge can only be inferred. Ricardo made up a series of fuels of different anti-knock values, each one being just on the point of detonating at a certain degree of supercharge in the sleeve-valve engine. Each of these different fuels was then tested for H.U.C.R. in the E. 35 engine under the usual conditions, and, from the known correlation between Octane numbers and H.U.C.R. in the E. 35 engine, the quality of the fuel required for each degree of supercharge could be stated in these terms if required.

Two series of tests were made, the induction air temperature being constant throughout a series, 25°–30° C. in the first and 75°–80° C. in the second. The air was supplied by an independent compressor, not driven by the engine under test, and since the temperature was kept constant throughout a series, it did not vary with the supercharge as it would in practice. The lower temperature in the first

series corresponds to a supercharge of about 2 lb. per sq. in. above atmospheric pressure, assuming 65 per cent. as the compressor efficiency, and the higher temperature to one of 8 lb. per sq. in.

The results of these tests are illustrated in fig. 26, in which the H.U.C.R. values and Octane numbers are given for the fuels just detonating at a compression of 4·3 : 1 with different degrees of supercharge. It is noteworthy that in these experiments no appreciable distinction can be drawn between the high and low temperature series.

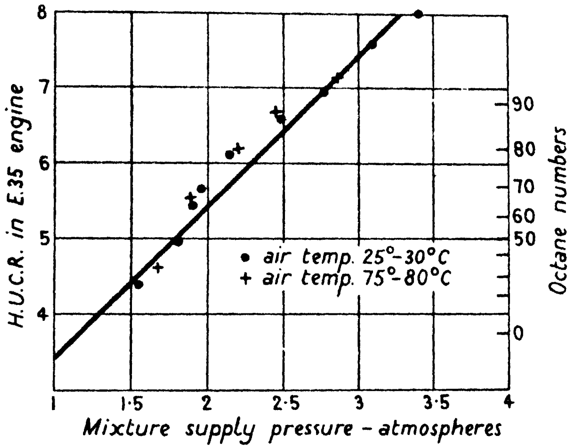

FIG. 26. Anti-knock values of fuels, expressed both in H.U.C.R. and Octane numbers, which just detonated with varying degrees of supercharge in an experimental sleeve-valve engine of C.R. 4·3. Speed, 1,500 r.p.m.

Reading from fig. 26, one can say that an increase of induction air pressure from 1·5 to 2·5 atm. had to be accompanied by an increase of H.U.C.R. in the fuel from 4·4 to 6·4, or, if the quality of the fuel be expressed in Octane numbers, from 26 to 84.

These H.U.C.R. values are for the E. 35 engine. We can arrive at an estimate of what the lowering of the compression limit in the supercharged engine would have been in the following way. We observe that a fuel with H.U.C.R. 4·4 withstood 1·5 atm. induction pressure in the sleeve-valve engine, and by producing the line backwards we can estimate that at 1·0 atm. (i.e. normal aspiration) a fuel of H.U.C.R. 3·4 in the E. 35 engine would have sufficed. In other words, a fuel which would detonate at 3·4 in the E. 35 would be of the same quality as one detonating at 4·3 : 1, in the sleeve-valve engine, both normally aspirated.

Owing to the absence of any hot exhaust valve, and to the favourable combustion chamber shape and sparking-plug location, it is

always found that a given fuel will stand a higher compression ratio in a sleeve-valve than in a poppet-valve engine, and the difference of rather less than unity in the compression ratio deduced from these tests has been confirmed by direct observation on other engines and at varying compression ratios. If we assume that this difference of 0·9 in the compression ratio between the poppet-valve and sleeve-valve engine would hold at other compression ratios, we can then argue that the fuel which detonated at 2·5 atm. in the sleeve-valve engine (Octane number 84) would have had an H.U.C.R. in that engine, normally aspirated, of 7·3; and similarly, that the H.U.C.R. of the one which detonated at 1·5 atm. (Octane number 26) would have been 5·3 in the same engine when normally aspirated.

We therefore arrive at the simple relationship shown in table 12, between the lowering of the allowable compression ratio and the supercharge measured in atmospheres, the one being twice the other.

It should be noted that throughout this series of experiments the anti-knock values of the fuel were adjusted by the addition of benzol. As mentioned at the beginning of this article a fuel with added benzol shows up badly as compared with a straight run fuel with lead ethide, if cylinder temperatures are high. It is very possible, therefore, that with a lead-doped fuel the necessary lowering of the compression ratio would not be so great as that shown in table 12.

TABLE 12

Lowering of allowable compression ratio in a supercharged engine.

Induction pipe pressure atm.	*Compression ratio allowable in sleeve-valve engine*		*Supercharge above atmosphere*	*Lowering of ratio below that when normally aspirated*
	With fuel of 84 Octane	*With fuel of 26 Octane*		
1·0	7·3 (estimated)	5·3 (estimated)	0	..
1·5		4·3 (measured)	0·5	1
2·5	4·3 (measured)		1·5	3

These figures of table 12, deduced from Ricardo's sleeve-valve experiments, have received satisfactory confirmation in subsequent work by Mucklow[3] using a single aero-engine cylinder of the usual poppet-valve type, and from experiments in the U.S.A.[4] In Mucklow's engine pinking first became audible at C.R.s 4·5 and 3·5, at induction pipe pressures of 1·07 and 1·54 atm. respectively. The same fuel, of aviation quality, was of course used in both cases.

Here again, a drop of unity in the compression ratio, accompanied

by approximately half an atmosphere of supercharge pressure, produced the same condition in regard to detonation.

Mucklow's experiments were made without heating the inlet air to correspond with the degree of supercharge, and in Ricardo's sleeve-valve cylinder the heating, when it was included, appears to have made little difference. The results given by Taylor,[4] based on experiments with a poppet-valve single-cylinder water-cooled engine, are valuable as supplementing those of Ricardo and Mucklow. Here again the conditions did not correspond with practice, because the

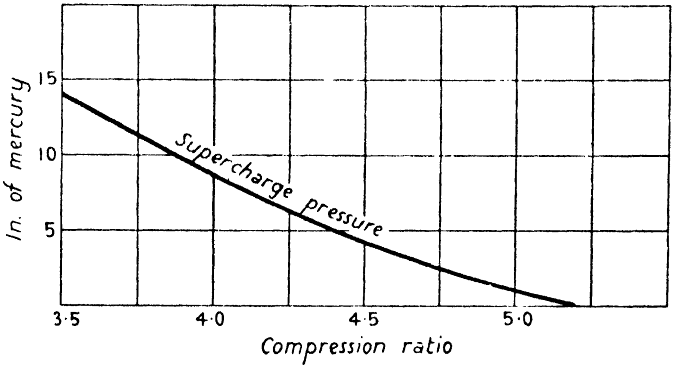

FIG. 27. Relation between compression ratio and supercharge pressure, with air temperature 65° C. Speed 1,510 r.p.m. Ignition adv. 35 deg.

inlet air temperature was maintained all the time at 65° C. instead of being raised with the supercharge pressure; but when compared with Mucklow's results, those of Taylor, illustrated in fig. 27, serve to show up the effect of a high air temperature in a poppet-valve engine.

It will be seen that a reduction of the compression ratio from 5·2 to 4·2 allowed an increase of the supercharge pressure by 7 in. of mercury or 3·5 lb. per sq. in., just half the amount allowable in Mucklow's experiments in which the air was not heated. The fuel-air mixture was sufficiently on the weak side to cause a reduction of 1 per cent. below maximum power.

It is not to be expected that the relationship between the degree of supercharge and the change of C.R. outlined above will hold over a wide range of compression ratios, and, as already pointed out, the relative effect of the two on detonation will probably vary with the type of fuel, and to some extent with the type of engine, so that the figures of table 12 and fig. 27 are only put forward as being roughly correct. They are useful, however, in helping towards getting a general picture of the behaviour of the supercharged engine.

ART. 19. *Heat-flow in a supercharged engine.*

The quantity of heat generated per cycle in the cylinder of a supercharged engine will be proportional to the charge weight of the fuel-air mixture supplied. If the thermal efficiency were the same as when normally aspirated, the total waste heat would go up in the same proportion, although not necessarily the part of it lost to the cylinder walls. But we have seen in the last article that with fuels liable to detonation it is necessary to work with a lower compression ratio, and therefore with a lower thermal efficiency, when supercharged. The total amount of waste heat will therefore increase more than *pro rata* with the weight of charge per cycle, when compared with what it might be with the same fuel, in the same engine normally aspirated, and with the compression raised to the limit for that particular fuel.

The points in a cylinder which are critical, in the sense of being the most liable to overheat under supercharged conditions, are the piston, the sparking plugs, and the exhaust valves if these are of the poppet-valve type. Adequate cooling of these parts depends upon each one being able to get rid of its heat sufficiently quickly by conduction to the walls of the cylinder: the piston through its skirt and the piston-rings to the cylinder barrel; the valves to the valve-seats during their periods of rest, and via their stems to the valve-guides; and the sparking plugs to the walls of the cylinder-head, helped by more or less of direct cooling in the air-stream, according to the design of the engine.

The difficulty of the problem of keeping the cylinder head and barrel, and hence the critical points, adequately cooled under supercharged conditions will turn upon the proportion of the total waste heat communicated to them. Happily the conditions in this respect have proved somewhat more favourable than might be expected.

Having fixed the compression ratio permissible with the fuel available, and with the maximum supercharge it is proposed to employ, then, as stated above, the I.H.P. and the heat generated per cycle may be expected to be closely proportional to the charge weight. If we neglect for the moment the power required to drive the supercharger, or imagine it to be separately driven, then the B.H.P. of the engine should increase rather more than in proportion to the charge weight, because friction losses will not increase in proportion to the power output, and the mechanical efficiency of the engine (apart from the supercharger) will improve as the power increases.

It has been found experimentally that the heat communicated to

the cylinder walls does not increase quite in proportion to the charge weight, and therefore that it becomes a substantially smaller fraction of the B.H.P. as the degree of supercharge is increased. Table 13

TABLE 13

Heat-flow to cylinder head and barrel in single-cylinder sleeve-valve engine at speed 1,500 r.p.m. Bore and stroke $4\frac{1}{2} \times 5\frac{1}{2}$ inches. Compression ratio 4·3 : 1.

Supercharging tests with air approximately at room temperature.

Intake air pressure (atm.)	*Mean air temp. ° C.*	*Relative charge density*	*Heat to cooling water in h.p.* to cylinder head	to cylinder barrel	Total (H)	*B.H.P. observed*	*B.H.P. corrected*	*(H) / B.H.P. (observed)*
1	19·75	1	3·29	10·71	14·0	15·3	15·3	0·915
1·52	26·5	1·485	4·98	15·25	20·2	28·8	27·6	0·70
2·02	27·0	1·975	6·55	19·5	26·05	41·5	39·1	0·63
2·50	27·0	2·44	8·0	23·9	31·9	53·3	49·5	0·60

gives the results of some experiments by Ricardo with the sleeve-valve cylinder referred to in the last article. It will be seen, from the first and last columns of this table, that while the air-intake pressure was increased from 1 to 2·5 atm. the heat carried away by the cooling water fell from 0·915 to 0·60 of the observed B.H.P.

These experiments illustrate the complexity of the supercharging problem, for several corrections have to be applied to the experimental figures as they stand, if they are not to be misleading. The supercharger was separately driven, in the first place, and it must be remembered that owing to the pressure of the air-supply positive work is done by the piston during what would ordinarily be called the 'suction' stroke. Some of the increase of B.H.P. shown in table 13 is therefore to be credited to the supercharger and not to the engine at all.

It is true that there will always be this positive work done during the charging stroke in a supercharged engine, but then the engine would normally be driving its own supercharger. In these circumstances the positive work on the crankshaft during the charging stroke may be regarded as a partial recovery by the engine of the work spent in driving the supercharger; but in a research engine with a separate supercharger it cannot be allowed to stand as an 'unearned increment' upon the B.H.P.

A further difficulty about interpreting the results of table 13 as they stand arises from the fact that the engine was specially built to withstand research under high supercharge pressures, and necessarily had very heavy rotating and reciprocating parts. Friction losses were

therefore unusually high, the lost power measured by motoring the engine (the L.H.P.) being equivalent to a M.E.P. of 30 lb. per sq. in. Although under supercharged conditions the L.H.P. assumed a more normal proportion of the I.H.P., on the other hand, when the engine was normally aspirated, the high friction losses caused it to have the unusually low mechanical efficiency of only 75 per cent. at 1,500 r.p.m. In consequence the rate of increase of the B.H.P. in relation to the I.H.P., as the charge pressure was increased, was more rapid than it would have been in an engine with a more normal figure for the L.H.P.

In deriving figures for the I.H.P. there is, of course, some uncertainty as to how the L.H.P. varied when the engine was supercharged. The figures in table 14 have been calculated on the assumption that the losses were always equivalent to a M.E.P. of 30 lb. per sq. in., as found in a motoring test when unsupercharged. In this table the I.H.P. and thence the B.H.P. have been corrected for the positive work on the charging stroke, and the variation of the corrected horse-powers therefore shows the effect of the supercharging upon the compression-explosion-expansion cycle in the cylinder.

The last three columns of table 14 show how the I.H.P. and the

TABLE 14

Supercharging tests with air at room temperature. Horse-power corrected for positive work on induction stroke. I.H.P. calculated throughout assuming a constant mechanical loss equivalent to 30 lb. per sq. in. M.E.P.

Air-intake pressure (atm.)	*B.H.P. corrected*	*I.H.P. corrected*	*Heat to cooling water in h.p.*	*Relative charge weight*	*Relative I.H.P.*	*Relative heat to cooling water*
1	15·3	20·2	14·0	1	1	1
1·52	27·6	32·5	20·25	1·605	1·61	1·44
2·02	39·1	44	26·1	2·20	2·18	1·865
2·50	49·5	54·5	31·9	2·80	2·70	2·28

heat to the cooling water varied with the relative charge weight. They show that while the I.H.P. increased almost exactly in proportion to the charge weight, the heat lost to the cylinder walls increased rather less rapidly than this, and therefore less rapidly than the total heat passing through the cylinder per cycle.

It should be noted that the relative charge weights of table 14 are not quite the same as the relative densities of the in-going charge in table 13, because allowance has to be made for the compression of the residual gas in the clearance space, from atmospheric up to the changing supercharge pressure.

In the experiments of tables 13 and 14 the air-intake temperature was maintained approximately atmospheric, and before too much attention is paid to them the corresponding data of tables 15 and 16 must be examined.

In the last column of table 15 the total heat to the cooling water

TABLE 15

Heat-flow to cylinder head and barrel. Supercharging tests with heated air. Other conditions as in table 13.

Intake air pressure (atm.)	Mean air temp. ° C.	Relative charge density	Heat to cooling water in h.p. to cylinder head	to cylinder barrel	Total (H)	B.H.P. observed	B.H.P. corrected	(H) B.H.P. (observed)
1	19·75	1	3·29	10·71	14·0	15·3	15·3	0·915
1·50	66	1·295	4·6	14·3	18·9	25·6	24·5	0·74
2·00	72·5	1·69	6·25	18·7	24·95	36·7	34·5	0·68
2·50	80	2·055	7·7	23·3	31·0	47·0	43·8	0·66

is again shown as a fraction of the observed B.H.P. The result of the high air-intake temperature is to make the heat to the cooling water a larger proportion of the B.H.P. Instead of dropping to 0·60 of the B.H.P. at 2·5 atm. intake pressure, as in table 13, it only drops to 0·66. Comparison of the last columns of tables 13 and 15 shows that the actual amount of heat to the cooling water at corresponding pressures was not much affected by the highei intake temperature, and that it was a higher fraction of the B.H.P. and I.H.P. merely because of the reduction in power under the new supercharged conditions, with the air heated, as compared with the low temperature tests in which the charge density was higher. The effect of the higher temperatures on the relative charge weights at the various pressures is seen by comparing the fifth columns of tables 14 and 16.

In a supercharged engine the air will be delivered with something more than the heat of adiabatic compression unless special 'inter-coolers' are fitted. For a pressure ratio of 1·5 the adiabatic rise of temperature would be about 35° C. and the actual rise, through a supercharger of 65 per cent. efficiency, nearly 55° C. The second test in tables 15 and 16, therefore, for 1·5 atm. pressure, corresponds well with practical conditions.

To summarize the results of this test in round figures we may say that a 50 per cent. increase in air-intake pressure gives a 40 per cent. increase in relative charge weight; that the I.H.P. is increased by 45 per cent.; and the heat to the cooling water by 35 per cent.

For calculating the I.H.P. figures in the supercharged tests of

TABLE 16

Supercharging tests with heated air. Horse-powers corrected as in table 14.

Air-intake pressure (atm.)	*B.H.P. corrected*	*I.H.P. corrected*	*Heat to cooling water in h.p.*	*Relative charge weight*	*Relative I.H.P.*	*Relative heat to cooling water*
1	15·3	20·2	14·0	1	1	1
1·50	24·5	29·25*	18·9	1·40	1·45	1·35
2·00	34·5	39·1*	24·95	1·89	1·94	1·78
2·50	43·8	48·4*	31·0	2·36	2·40	2·22

* Lost h.p. taken as being the same as in test (1) with air-intake at room temperature.

table 16 a mechanical loss corresponding to a M.E.P. of 30 lb. per sq. in. has been taken, as before. The motoring loss on which that figure was based was measured with the air-intake at room temperature, and it is quite possibly an overestimate for the high temperature tests, when the oil on the cylinder walls would be less viscous. If the true friction loss in the last three tests of table 16 was equivalent to 25 instead of 30 lb. per sq. in., then the calculated I.H.P.s come out *exactly* proportional to the weights of charge.

The results of supercharging upon the B.H.P., I.H.P., and heat to the cooling water, as set out in tables 13 to 16, have been collected together in fig. 28 in which the relative increase of each quantity is shown plotted against the relative weight of fresh charge, as compared with the normally aspirated condition. The full line curves are drawn to show the results of the lower temperature tests of series (1)—tables 13 and 14—while the dotted curves and crosses show, by their distances from the full line curves, the effect of the raising of the charge temperature by 40°–50° C. in series (2).

There are signs of a slightly more rapid increase of B.H.P. with the charge weight in series (2), due probably to lower friction losses. Since the same friction loss has been assumed for each series this increase of B.H.P. is reflected in the I.H.P. As mentioned already, a reduction of 5 lb. per sq. in. in the assumed friction M.E.P. for series (2) would make the I.H.P. exactly proportional to the charge weight and bring the curves for the two series into coincidence.

It was pointed out earlier that when compared at the same supercharge pressure the rate of heat-flow to the cooling water was not much altered as between series (1) and (2). When compared at the same relative charge weight, however (and therefore with the same heat generated per cycle), the effect of the higher average gas temperature is seen in the greater steepness of the dotted curve through the observations of series (2). For the same reason the crosses lie appreciably above the lowest curve in the figure.

One cannot leave the account of this very difficult experimental investigation by Ricardo without paying a tribute to the extraordinary accuracy and consistency of the measurements made; for it may be added that the necessary corrections to the original observations, and other calculations upon which tables 13–16 are based, have been made independently and some considerable time after the work was done.

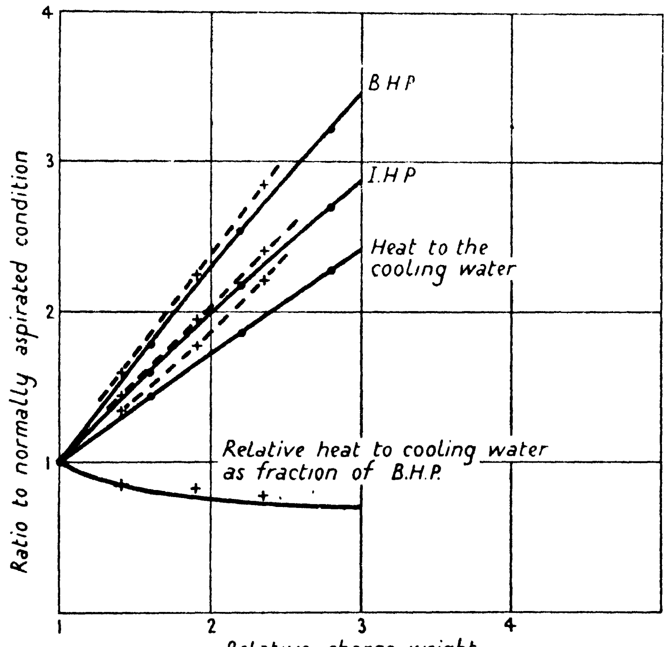

FIG. 28. Relative B.H.P., I.H.P., and heat to the cooling water, for various degrees of supercharge. Single-cylinder engine. C.R. 4·3. Speed 1,500 r.p.m.

ART. 20. *Fuel-air ratio in a supercharged engine.*

With some variation, depending upon the fuel and the compression ratio, the range of fuel-air ratios over which steady running is possible in a normally aspirated engine may be taken broadly as extending from 15 per cent. weak to 50 per cent. rich. In a supercharged engine it is always found that this mixture range is reduced. It may be found that unsteady running sets in before the normal limits are reached either at the weak or the rich end of the range, or at both. This depends upon the adjustment of the ignition timing and upon the compression ratio. But it has been the experience of all investigators that the workable range becomes reduced as the induction pipe pressure is raised.

The most comprehensive series of tests on the subject are those of Mucklow[3] on two different types of aero-engine cylinder, at compression ratios varying from 3·5 to 7 : 1, and with induction pipe pressures from 1 to nearly 2 atmospheres. Some of this series of tests—at the lower ratios—were carried out with a fixed ignition

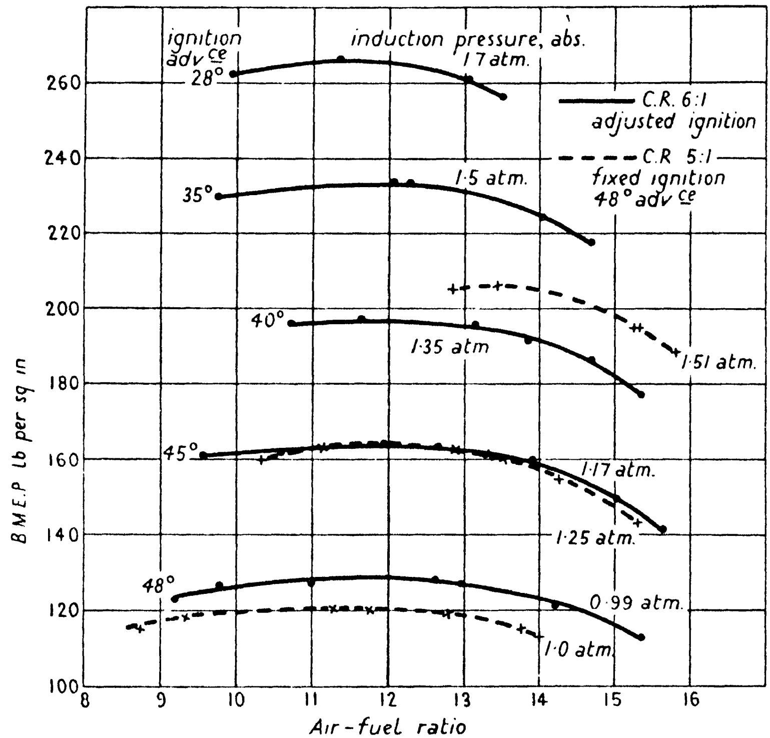

FIG. 29. Variation of mixture range in a supercharged engine. Speed 1,600 r.p.m.

timing, while in others the timing was varied so as to give maximum power under each set of conditions. Under either condition the mixture range was reduced, but the actual limits were affected by the ignition timing in the manner illustrated by the typical curves shown in fig. 29.

A summary of Mucklow's results at all compression ratios, when using benzol as fuel, is given in table 17. From this it will be seen that the narrowing of the mixture range occurred at all ratios, and whether or not the ignition timing was varied. So extreme was the limitation of the range at 4 : 1, with fixed ignition timing, that when supercharged to 2 atm. it was impossible to work upon the rich side of the correct mixture at all. This feature, i.e. a shifting of the work-

TABLE 17

Variation of mixture range with induction pipe air pressure. Speed 1,600 r.p.m. Fuel, benzol. Air-fuel ratio for correct mixture 13·6 : 1.

Compression ratio	*Induction pipe pressure (atm.)*	*Ignition advance degrees*	*Permissible air-fuel ratio*	
			Weakest	*Richest*
4	0·99	48·5°	13·1	8·8
	1·24	”	14·0	10·7
	1·47	”	14·3	11·8
	1·70	”	14·8	13·2
	2·0	”	15·7	14·3
5	1·0	”	14·0	8·8
	1·25	”	15·3	10·3
	1·51	”	15·8	12·8
6	0·99	48·5°	15·4	9·2
	1·17	45	15·7	9·6
	1·35	40	15·4	10·7
	1·50	35	14·7	9·7
	1·70	28	13·5	9·9
7	0·99	40°	16·2	9·3
	1·20	35	15·2	9·5
	1·37	28	14·7	10·3

able range towards the weak end, was always found so long as the ignition timing was fixed: it is shown by the dotted curves in fig. 29. With adjustment of the ignition timing the ranges, while still shortened under supercharged conditions, retained the symmetrical arrangement shown by the full lines in the same figure.

It can be seen from table 17 that in the normally aspirated condition there was a progressive weakening of the limiting mixture, as the compression ratio was raised, from 13·1 at 4 : 1 to 16·2 at 7 : 1. This feature of the results is in accordance with the effects of a change of ratio upon combustion as set out in art. 36 (i).

With varied ignition timing, which may be taken as representing normal practice, the limit on the rich side is seen in table 17 to lie always between 9 and 10 : 1 corresponding to a mixture about 50 per cent. rich, and the narrowing of the range was due to reductions in the weakest permissible ratio. It is to be noted that according to Mucklow's figures his engine even when normally aspirated would not run steadily on a mixture more than 11 per cent. weak at C.R. 7 : 1 and not more than 3 per cent. weak at C.R. 5 : 1. Such limits are difficult to account for unless there was some leak of air into the induction system other than through the measuring orifice.

The conclusions of the whole investigation, which are substantially

confirmed by the work of Ricardo and others, may be summarized by saying:

(1) that in all circumstances the workable mixture range is narrowed by supercharging;

(2) that an increase of compression ratio tends to extend the mixture range on the weak side, while leaving it unaffected on the rich side;

(3) that with fixed ignition timing there is a tendency for the working range to be shifted bodily, by supercharging, in the direction of the weak mixtures;

(4) that this shift towards weakness is eliminated by adjusting the ignition to its optimum value for each set of conditions.

ART. 21. *The practical limits of power output per unit of swept volume.*

Having analysed, in the foregoing articles, the basic conditions which control the behaviour of a supercharged cylinder, it is now time to summarize the position reached in present-day practice. This is limited to the use of the gear-driven supercharger for maintaining an induction manifold pressure within a few lb. per sq. in. of normal atmospheric, some extra boost being allowed temporarily during the take-off.

In considering how the maximum B.M.E.P. of any cylinder may be increased, the three main variables are compression ratio, supercharge pressure, and fuel quality; but the problem in practice centres upon the fuel. The need for a good thermal efficiency means that the compression ratio must lie in the region 5·5–6 and the struggle for a high power output then becomes a question of obtaining a fuel which will stand an extra lb. per sq. in. of supercharge without detonation. A well-designed combustion head will assist, but the principles of good design are now generally understood, and with some variability according to the size, design, and method of cooling of the cylinder, one can give a direct relation between the anti-knock value, expressed in Octane number, of the fuel, and the maximum B.M.E.P. which can be maintained without detonation. This has been done by Taylor[4] who shows a linear increase of the B.M.E.P. from 130 to 200 for an increase of Octane number from 65 to 95.

The range of B.M.E.P. appears high, for there can be few engines capable of sustaining a B.M.E.P. of 130 lb. per sq. in. on a 65 Octane fuel. Details are not given, but one may probably conclude that the figures refer to a water-cooled engine, and possibly with the air supply not heated as it would be by a centrifugal supercharger. A similar relationship found for an air-cooled engine at 2,200 r.p.m. is shown in fig. 30. In these tests the C.R. was 5·6 to 1 and the air

temperature 70° C., so that the conditions were as severe as they are ever likely to be in practice. The tests were made by increasing the boost pressure, at the constant air temperature of 70° C., until incipient detonation was observed with each fuel.*

The rise of permissible B.M.E.P. shows the same linear increase as was given by Taylor, but the rate of increase is rather higher. The lower range of B.M.E.P.s may very well be due to the high air temperature and to the tests of fig. 30 being upon an air-cooled

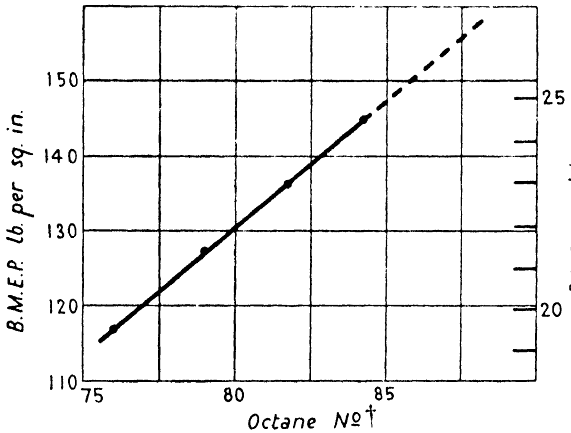

FIG. 30. Increase of possible B.M.E.P. with the anti-knock value of the fuel used. Air-cooled cylinder 5¾ in. × 7½ in. Speed 2,200 r.p.m. C.R. 5·6. Air temp. 70° C. Ignition adv. 30 deg. Consumption 0·55 lb. per B.H.P. hour.

cylinder. The constant air temperature would also tend to exaggerate the apparent increase of M.E.P. with Octane number, by depressing the power obtained with the inferior fuels at zero boost below what could be maintained by a normally aspirated cylinder.

At the right-hand side of fig. 30 there has been added a scale of B.H.P. per unit of swept volume, for the air-cooled cylinder at 2,200 r.p.m. The four observations cover a range from about 19 to 25 h.p. per litre, which compares, it will be remembered, with 64 in the Schneider Trophy engines of 1931 and with 107 in Ricardo's small sleeve-valve cylinder.

In a normally aspirated engine the maximum torque obtainable must vary with the speed, on account of restrictions at the valves, and the maximum permissible speed of rotation may not, therefore, be that which gives maximum power, if there should be a bad falling

* F. R. Banks, in a recent paper,[69] has given fuller information of the same kind.

† As determined according to the British Air Ministry fuel specification No. D.T.D. 230.

off in volumetric efficiency before it is reached. In an engine with a gear-driven supercharger, on the other hand, the maximum torque near the ground is independent of the speed, for the engine must in any case be throttled, and the torque can be raised, at the maximum permissible speed, to whatever point detonation and the proper cooling of the cylinders allow.

About 6 per cent. of the gross power output will be absorbed in driving the supercharger, and each step up in the boost pressure which the anti-knock quality of the fuel may permit means a rather more than proportional increase in the net B.H.P.

The mean pressures shown in fig. 30 are the gross values, obtained with a separately driven supercharger, and the figure of 22 B.H.P. per litre obtained with 80 Octane fuel would correspond to a net figure of about 20·5 in a complete engine. This is equivalent to 34 h.p. per 100 cu. in., and is fairly representative of the output of modern engines with cylinders of 5 in. bore and over. For small cylinders of about 3½–4 in. bore and much higher rotation rates, a figure of 50 h.p. per litre seems to be well within sight.

These figures of B.H.P. per litre refer to engines of the necessary life and reliability for flight, and capable of withstanding the 100 hour type test referred to in Chapter I. They must not, therefore, be compared with the extreme figures realized in some racing-car and motor-cycle engines, where the same length of working life is not required.

The maximum B.H.P. obtained from a given cylinder without detonation or overheating must, of course, depend largely upon the fuel-air mixture employed. There is very little drop in power even when a mixture is made 50 per cent. rich, and by allowing such an excess of fuel to help to cool the cylinder, powers can be maintained with a rich mixture which would involve heavy detonation with an economical one. The relation between cylinder temperatures and mixture strength will be considered more fully in Chapter XIII, and all that need be included here is to show how the maintenance of power and fuel economy are both related to a high anti-knock quality in the fuel. This is illustrated in figs. 31 and 32, both taken from Taylor's[4] paper, in the first of which the power output of a Wright Cyclone engine is plotted against fuel consumption per B.H.P. hour for five different qualities of fuel. With the inferior fuel of 65 Octane number it was impossible to maintain the full power of 600 B.H.P. below 0·62 lb. per B.H.P. hour, whereas with the 82 Octane fuel there was no rapid fall off in power until the fuel consumption had been reduced to 0·53.

In fig. 32 the effect of an inferior fuel, and the resulting detona-

tion, upon cylinder temperature is exhibited for the same five fuels, also plotted against the rate of fuel consumption per B.H.P. The rapid rise of temperature which accompanies heavy detonation will be noticed in the case of the two worst fuels.

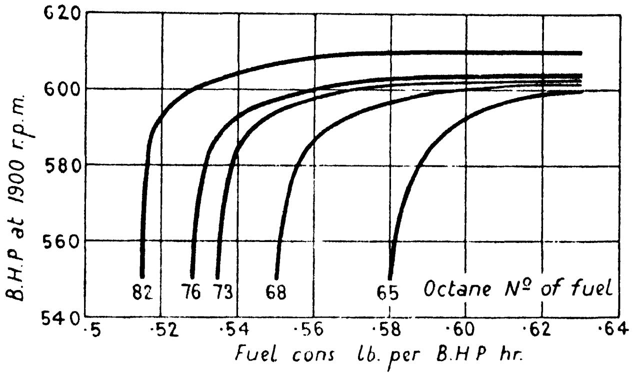

Fig. 31. The effect of the quality of the fuel upon engine power output.

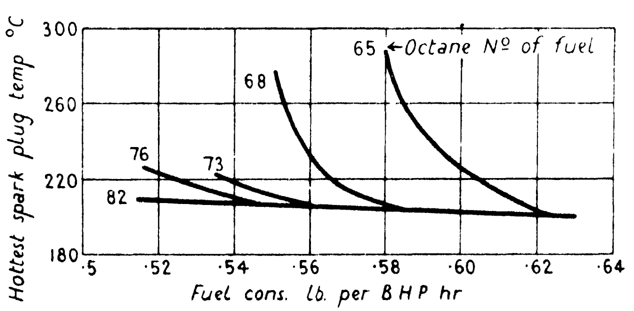

Fig. 32. The effect of the quality of the fuel on engine temperature.

The bore and stroke of the Cyclone engine being $6\frac{1}{8} \times 6\frac{7}{8}$, and the displacement 1,820 cu. in., 600 h.p. at 1,900 r.p.m. is equivalent to 138 B.M.E.P. and 20 h.p. per litre. At 0·55 lb. per B.H.P. hour the engine just failed to give this output with a fuel of 76 Octane, and the tests in fig. 30, made at the same fuel consumption per h.p. hour, exhibit very nearly the same result.

IV

PISTON AND VALVE TEMPERATURES

ART. 22. *The functions of the piston.*

A critical factor, perhaps *the* critical factor, in the operation of any high duty engine is the piston; and the many problems connected with it nearly all centre round the question of heat-flow. The primary duty of the piston is to act as a gas-tight plunger which transmits the gas forces to the crankshaft; but in so acting it cannot avoid absorbing a considerable amount of heat, and its secondary duty, therefore, is to provide a passage of escape for a portion of the waste heat which must ultimately find its way to the cylinder walls or to the lubricating oil.

In large cylinders of 12 in. diameter and over, elaborate arrangements have to be made to circulate a supply of cooling water to and from the piston through the moving connecting-rod. In smaller sizes the circumference bears a larger ratio to the piston surface over which heat is received from the burning gases, and it is possible to provide adequate cooling by conduction to the cooler cylinder walls.

Fig. 33 illustrates an aero-engine piston of normal design, with two 'gas rings' and two 'scraper rings', of which more anon. The primary function of the gas rings is to maintain the gas-tight seal for the working substance above the piston, but they perform, also, the no less important function of helping to convey heat from the piston to the cylinder walls. Between the piston body and the cylinder wall there must always be a working clearance, of an amount depending upon the materials they are made of but which is roughly 0·003 in. per in. of piston diameter for an aluminium alloy piston when cold. The clearance space, even under working conditions, will be filled with a film of oil some thousandths of an inch thick, and the rate of conduction of heat through this oil film would be too slow to provide adequate cooling of the piston without assistance from the rings. It has been suggested that the piston-to-cylinder contact is a case of 'boundary lubrication' as described in art. 29. Probably that is so over a very small part of the circumference; but Ricardo and others have carried out experiments which prove that the total piston friction is closely proportional to the viscosity of the lubricant, and this could only be so if the oil film over most of the piston surface were of an appreciable thickness. Piston rings are designed to exert a constant uniform pressure all round the cylinder wall, and the thickness of the oil film between their outer surface and the cylinder is no doubt

far less than elsewhere and may on occasion be of molecular dimensions, namely of the order of 10^{-8} in. This minimum thickness of oil film, however, could only be maintained between a perfectly circular cylinder and ring, and provided the edges of the ring were sharp enough to scrape off all surplus oil, down to the primary layer required for boundary lubrication. No cylinder or ring surface is perfectly cylindrical, judged in terms of molecular dimensions, and in a high duty engine, more particularly with an air-cooled cylinder, slight distortion of the cylinder barrel is bound to occur on account of unequal temperature distribution. This and the local unevenness

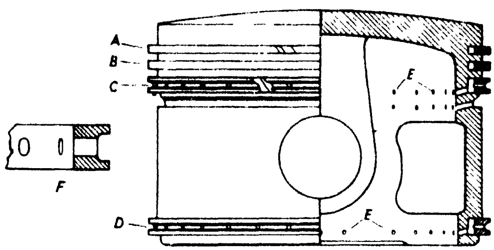

FIG. 33. Typical aero-engine piston.

of surface prevents the perfect fitting of rings in the cylinder, and results in an oil film of finite and varying, though very small, thickness, even under the rings. When the distortion of the cylinder is serious it may give rise to excessive oil consumption and even to complete failure of the piston.

The primary duty of the gas rings, to prevent hot gas from getting down the sides of the piston, is vitally important for reasons beyond the mere need of avoiding gas leakage and loss of the working pressure on the piston. At high speeds this would always be negligible. The very life of the engine, however, depends upon the tightness of the gas rings, because a leakage of the hot gases down the sides of the piston immediately begins to destroy the lubricating properties of the oil, turning it, in fact, first to a gummy consistency and then to hard, caked carbon. The rapidity with which disaster may follow such a failure of gas tightness is astonishing. An engine normally capable of running for hundreds of hours may be wrecked by ten minutes with a leaky piston ring. The sequence of events is usually that a ring first becomes stuck in its groove by contamination and gumming of the oil. Once stuck, the ring is unable to accommodate itself to the changing thrust of the connecting-rod and to the consequent side-to-side movement of the piston; it no longer fits closely to the cylinder wall and is now useless alike for conducting heat from

the piston and for maintaining gas tightness; the piston heats up, destruction of the oil is accelerated, other rings become stuck, and around them, and between the piston body and the cylinder, carbon begins to form. When this stage has been reached nothing but an immediate stopping of the engine can prevent complete seizure.

Some alloy of aluminium, commonly in the forged condition, is now universally used for pistons in high duty engines for two reasons: its lightness reduces the inertia forces on the bearings at high speeds, and it has a specific conductivity for heat more than four times that of cast iron. With the first point we are not concerned at present, but the second is of primary importance in its effect upon the power output of an engine. When discussing detonation in vol. i it was emphasized that this is promoted by the presence of hot spots in the walls of the combustion chamber, and the two most difficult points to cool are the exhaust valve and the centre of the piston. Since all the heat leaving the piston has first to be conducted through the piston crown, either axially to the under surface from which it is carried away by the lubricating oil, or radially until it reaches the rings at the periphery, the temperature at the centre will be directly affected by the thickness of metal and its conductivity, in a manner to be dealt with in art. 24. On both counts the aluminium alloy piston scores heavily, and its introduction was the sign for an immediate increase in maximum cylinder output on account of the much greater ease of keeping it reasonably cool.

ART. 23. *Piston temperatures.*

Although direct measurements of the working temperatures of a piston have been successfully made by means of embedded thermocouples on slow-speed engines, notably by Hopkinson,[5] Coker and Scoble,[6] Riehm[7] and others, and even up to 800 r.p.m. by Jardine and Jehle,[8] it would be an almost hopeless task to attempt direct observations on an engine designed to run at 2,000 r.p.m. The technique of measuring piston temperatures in such an engine is that due to Gibson.[9] Although observations are not actually made while running, there is no doubt that the running temperature can be obtained, and the observations repeated time after time, to within an accuracy of $\pm 3°$ C.

The method consists in drilling a series of small holes into the piston top at different radii; running the engine at a given load under steady conditions; then, at a given instant, closing the throttle, stopping the engine quickly by applying a brake, removing a sparking plug, and passing a thermocouple through the plug-hole and inserting it into one of the holes drilled in the piston to receive it.

Thereafter a cooling curve is taken, from which the piston temperature before the engine was stopped can be inferred.

It was found by Gibson that with practice a first reading could be taken within 10 sec. after the closing of the throttle, and in a more recent development of the method by Wright Baker[10] this interval

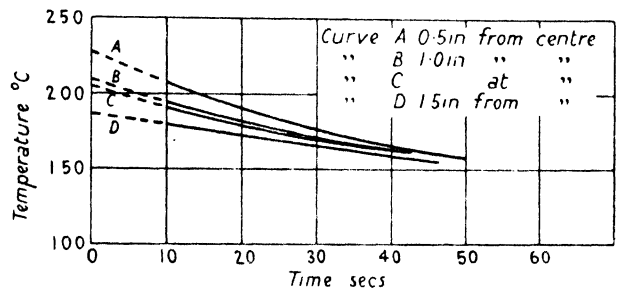

FIG. 34. Cooling curves for an aluminium piston of 100 mm. diameter.

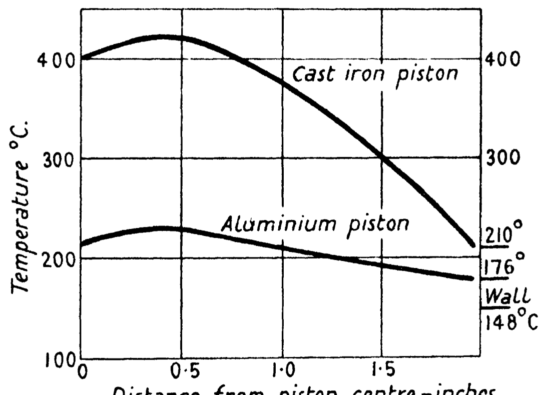

FIG. 35. Working temperatures of an aluminium and a cast-iron piston each of 100 mm. diameter.

was reduced to 5 sec. Owing to the high temperature of the surrounding cylinder walls and valves, the cooling of the piston is sufficiently slow to allow an accurate extrapolation of the cooling curve backwards, and the working temperatures of the piston can be inferred with certainty to within about 3° C.

Typical cooling curves given by Gibson for an aluminium piston of 100 mm. diam. (3·94 in.) are reproduced in fig. 34. The dotted portions of the curves show the necessary extrapolations to find the piston temperatures at the moment of closing the throttle. Values of these working temperatures at different distances from the piston centre, as deduced from the cooling curves, are plotted in fig. 35.

These observations were made with an air-cooled cylinder, and the observation points were along a radius of the piston towards the cylinder wall in the down-stream direction. It will be noticed that the hottest point was not at the piston centre, but was displaced towards the hottest part of the cylinder wall.

In fig. 35 the temperature curve is also given for a cast-iron piston of the same diameter. The much higher range of temperatures on this piston follows from its lower thermal conductivity, 0·102 C.G.S. units at 350° C. as against 0·38 C.G.S. units for the aluminium alloy at 200° C., and from the greater thickness of the crown of the aluminium piston, 6·5 mm. as compared with 3 mm. for the cast iron. In spite of its thicker cross-sections the weight of the aluminium piston was 1·262 lb. as compared with 1·776 lb. for the cast-iron one.

The leading features for the two pistons are summarized in table 18. The tests were carried out under comparable conditions, except

TABLE 18

Comparison of the temperatures of an aluminium alloy and a cast-iron piston of the same diameter, 100 mm., and the same general design, except that the crown of the C.I. piston was 3·0 mm. thick as compared with 6·5 mm. in the aluminium one. Speed 1,800 r.p.m.

	Aluminium	*Cast iron*
Maximum temperature	228° C.	420° C.
Temp. fall, hottest point to periphery	52°	210°
Temp. drop, periphery to wall . .	28°	62°
Wall temperature	148°	148°

that, owing to its high temperature, the B.M.E.P. with the cast-iron piston was only about 112 lb. per sq. in. as compared with 120 in the case of the light alloy one. The petrol-air ratio in all the tests was the weakest one which would give maximum power, this being the fuel-air mixture which was found to produce the hottest piston.

The important features from the point of view of data of general applicability are the relations between the maximum and the peripheral piston temperatures, and the relation between the latter and the temperature of the cylinder wall. In Gibson's 100 mm. piston the edge was 52° C. below the maximum temperature and only 28 C. above the wall temperature on the hot side of the cylinder. In Jardine and Jehle's experiments upon a 5-in. Liberty engine piston with a crown 9·5 mm. (⅜ in.) thick, and thermocouples at the points shown in fig. 36, there was a difference of about 50° C. between the hottest point, in this case the centre, and the point marked 2 at a radius 1⅝ in. The rate of fall of temperature towards the edge was

therefore much the same, although the greater ratio of surface to periphery in the larger piston had raised the maximum to about 350° C.; and this in spite of its working in a water-cooled cylinder, and at 800 instead of 1,800 r.p.m.

A new and important feature was demonstrated in Jardine and Jehle's experiments which was not brought out in those by Gibson, namely, the importance of the piston rings in keeping the piston

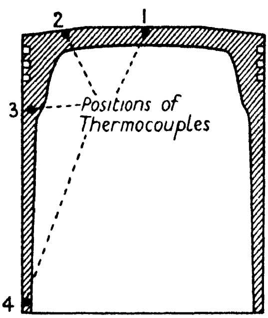

Fig. 36. Section of piston, showing points of temperature measurement in Jardine and Jehle's experiments.

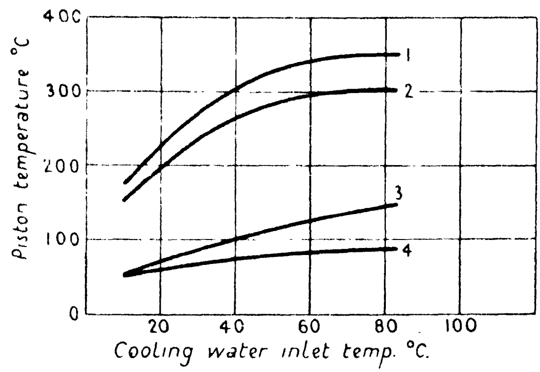

Fig. 37. Temperatures of Liberty engine piston at the points 1, 2, 3, and 4, shown in fig. 36, for different temperatures of the cooling water. Cylinder bore and stroke 5 in. × 7 in. Speed 800 r.p.m. B.M.E.P. 95 lb. per sq. in.

cool. The results in fig. 37 are shown for different inlet temperatures of the cooling water. If one considers those for a water temperature of 80° C. and a wall temperature, therefore, of 85°–90° C. it will be seen that the temperature in the piston crown $\frac{7}{8}$ in. from the periphery (curve 2) was no less than 300° C. Only a little way below the rings, on the other hand (curve 3), the temperatures were down to well below the peripheral temperatures in Gibson's experiments.

The importance of the rings and ring-belt in the cooling of the piston crown has been confirmed by Baker's experiments referred to above. These covered a large number of pistons of widely different types, all tested in the same water-cooled cylinder, of bore and stroke $3\frac{1}{4}$ in. × $5\frac{1}{4}$ in., at the same speed of 1,640 r.p.m. The results confirm those of Gibson so far as the measurement of the temperatures in the piston crown are concerned. Over the whole range of normal types the temperatures at the centre lay between 220° and 240° C. for the aluminium alloy pistons, and were 450°–500° C. in the cast-iron ones. The mean temperature difference between the edge of the crown and the cylinder wall, for 21 different aluminium pistons, was 84° C.

From temperature measurements in which one or more rings were removed, Baker concluded that not more than about 10 per cent. of the total heat received by the piston was dissipated from the skirt; and that the lands between the rings, with minimum hot clearances, were even more effective, area for area, than the rings themselves. Furthermore, from experiments made with baffles on the underside to protect the piston from oil-splash and air turbulence, he concluded that only about 10 per cent. was carried off in that way from his aluminium alloy pistons, leaving about 80 per cent. as the portion transferred to the cylinder walls either via the rings or from the lands above the bottom ring.

Information is not available as to the temperatures of aluminium alloy pistons of 5–6 in. bore, working in cylinders which give the extreme figures for h.p. per litre now reached in some aero-engines. In these the maximum of about 240° C. indicated above would no doubt be exceeded. Mr. Baker kindly allows me to quote some figures from his latest work* which show that in a $3\frac{3}{8}$ in. piston the maximum temperature rose from 220° to 262° C., with the same mean temperature of the (air-cooled) cylinder, 140° C., when the h.p. per litre was doubled by an increase of speed, from 17·4 at 2,000 r.p.m. to 35·6 at 4,000 r.p.m.

Differences of size, of design, and of the power output of the cylinder per litre, must have a substantial effect on piston temperatures, so that close estimates are impossible, but the following broad conclusions from the results given above would appear to be justified:

(1) For pistons of 5 in. diameter and more, and whenever there is difficulty in getting rid of the waste heat, the rings and ring-belt play a very important part, and are able to reduce the temperature at the top of the skirt to about 60° C. above the wall temperature.

(2) In smaller pistons, where the maximum temperatures are less than 250° C., there is not always the same large drop of temperature across the ring grooves, because even the temperature at the edge of the crown can be kept down to about 80° above the wall temperature.

(3) The observed effectiveness of the lands in dissipating heat suggests that the heat is transferred between the piston and cylinder surfaces by convection in the oil film, and the rate of transfer will, therefore, increase with the mean piston velocity. This would explain the quite moderate rise of piston temperature mentioned above when the power output was doubled at twice the speed.

(4) Under no conditions is it important to provide a large area of skirt from the point of view of heat transmission.

This last point receives confirmation from experiments by Gibson

* To be published in *Proc. I.A.E.*, vol. xxix.

in which the temperature of the same piston was measured with widely different skirt clearances. From the original of 0·024 in. (measured cold) on the 4-in. diameter piston, the clearance was increased to 0·045 in. The comparative piston temperatures are shown in fig. 38. The mean wall temperatures on the hot side are also given. It will be seen that the increased clearance produced an increase of the drop in temperature between the piston and the

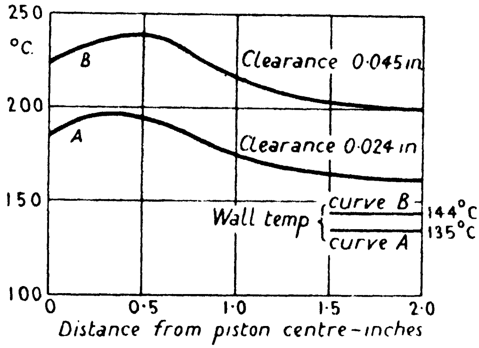

FIG. 38. Temperature curves for an aluminium piston, 100 mm. diameter, with different values of the skirt clearance.

wall from 27° to 56° C. The temperature of the hottest point was increased by 42°, but since the wall happened to be 9° hotter with the larger clearance, the fall of temperature from the hottest point to the edge is nearly the same in each case.

Since an increase of the normal clearance by 90 per cent. produced a rise of the maximum temperature of only 42° C., and of the drop at the wall of 29° C., it is safe to conclude that any ordinary variation of clearance likely to be met with in practice will not have an important influence upon piston temperature.

In some further experiments the area of the piston skirt was reduced by 12 per cent., from 33·4 to 29·4 sq. in., by cutting holes where strength was not required, and this produced an increase of the temperature drop at the wall from 26° to 39° and an increase of the maximum temperature of only 11°, from 197° to 208°. Although the increase of temperature was slight, so also was the saving of weight, being only 0·1 lb. in a total of 1·408 lb. The general practice in modern designs of light pistons was illustrated in fig. 33, from which it will be seen that the skirt is scarcely deeper than is necessary to provide for the gudgeon-pin bosses, but no attempt is made to lighten it further by the removal of metal. At the ends of a diameter at right angles to the gudgeon-pin an adequate surface is required to carry the side thrust on the cylinder wall.

As regards the other variable features of design, such as the presence or absence of ribs beneath the piston crown, it was found in Gibson's experiments that the temperature difference between the wall and the piston varied only between 25° and 28° over all the types tried. The temperature at the hottest point was found to depend mainly, as one would expect, upon the thickness of metal in the crown. Gibson concluded, from his experiments, that the advantage, for a given size and weight of piston, lay in doing away with the ribs and putting the metal saved into any necessary increase of thickness of the crown. The same conclusion, which, however, needs qualification as indicated below, was reached by Jardine and Jehle. They found that a reduction of thickness in the crown from $\frac{3}{8}$ in. to $\frac{1}{4}$ in. increased the temperature drop between points 1 and 2 in fig. 36 from 50° to 75° C.

The conclusion that metal can more usefully be put into a general thickening of the piston crown than into a system of ribs and a less thick crown, may require to be modified in an engine in which oil-splash plays a large part in keeping the pistons cool; and also in an engine where very high gas-pressures make the strength of the piston crown a critical factor. In cooling by oil-splash, the extra metal surface provided by a system of small ribs may help the dissipation of heat from the underside of the piston.

Finally, it may be noted that between five different types of aluminium alloy tested by Baker, differences of conductivity had a negligible effect upon the maximum temperature, all the five falling within a range of 7° C.

ART. 24. *Heat-flow in the piston crown.*

The radial variation of the temperature of a flat circular disk which is receiving heat uniformly over one surface can be calculated in terms of this rate of heat reception, the thickness of the disk, and the thermal conductivity of the material. Hopkinson[5] showed that if one treats the crown of a piston as a disk of constant thickness t and conductivity k, which is receiving heat at the rate of h calories per sq. cm., then the difference of temperature between two points at radii r_1 and r_2 will be

$$\theta_1-\theta_2 = \frac{h(r_2^2-r_1^2)}{4kt}, \tag{12}$$

or, alternatively, if t varies with r, so that $t = cr$, then

$$\theta_1-\theta_2 = \frac{h(r_2-r_1)}{2kc}. \tag{13}$$

In these expressions h must be taken as the difference between the rate of reception of heat at the top surface and its dissipation from the underside of the piston. In some engines, as already mentioned, the amount of heat carried away by oil-splash from the underside of the piston may be very considerable.

With the aid of the above equations, and of the type of temperature measurements described in the last article, Gibson[11] has deduced the rate of dissipation of heat from the periphery of a number of different types of piston.

In table 19 are given the values found by Gibson from his own

TABLE 19

Rates of heat dissipation and receptivity deduced from experiments with various types of piston.

Authority	*Engine type and speed*	*Piston bore × stroke and material*	h *C.G.S. units*	e *C.G.S. units*
Hopkinson . .	Gas engine 180 r.p.m.	C.I. 11·5 in. × 21 in.	1·43	$3{\cdot}9 \times 10^{-6}$
Coker . . .	Gas engine 200 r.p.m.	C.I. 7 in. × 15 in.	1·16	$3{\cdot}4 \times 10^{-6}$
Jardine and Jehle	Aero-engine 800 r.p.m.	Aluminium 5 in. × 7 in.		
		(a) $t = 0{\cdot}25$	4·15	$9{\cdot}0 \times 10^{-6}$
		(b) $t = 0{\cdot}375$	4·05	$8{\cdot}8 \times 10^{-6}$
Gibson . . .	Aero-engine 1,800 r.p.m.	Aluminium 100 mm. × 140 mm.		
		(4 types)	5·56	$11{\cdot}0 \times 10^{-6}$
		„ (5th type)	5·75	$11{\cdot}0 \times 10^{-6}$
		C.I. 100 mm. × 140 mm.	1·60	$5{\cdot}1 \times 10^{-6}$

and from the other researches of which mention has been made, and also the values he deduced for the 'receptivity' e, defined as the rate of reception of heat per sq. cm. per ° C. difference of temperature between the piston and the gas. These values of e are calculated on the assumption that the mean value of h is not proportional to the mean temperature difference ϕ between the piston and the gas, but to the mean value of ϕ^2 throughout a cycle, so that

$$h = e(\phi^2)_{\text{mean}}.$$

The values given for e are of little practical use for finding the rate of heat loss to the piston, in the first place because it is difficult to estimate the mean square of the temperature difference throughout a complete cycle with any accuracy; and secondly because the value of e itself will certainly vary widely according to the gas turbulence in the cylinder, and is likely to be different therefore at every

speed, and to depend, in different engines, upon the number, size, and arrangement of the valves.

The differences of gas turbulence in the engines of table 19 are reflected in the figures for *e*, which, with the exception of Gibson's cast-iron piston, move up roughly with the speed. The very much lower receptivity of the cast-iron piston in the last line, than that of the aluminium ones at the same engine speed, is difficult to explain. The difference is attributed by Gibson partly to carbon deposit on the piston crown and partly to a high rate of heat loss from the underside of the piston through oil-splash; for it must be remembered that the piston was some 200° C. hotter than the aluminium ones, while the oil would be about the same temperature in both tests.

In the light of these uncertainties the figures given are clearly not of general applicability, but for engines of which the speeds, as well as the cylinder and valve-gear design, are generally similar to those to which the figures refer, they may probably be employed with safety. Gibson made use of the values of *h* deduced from the piston temperature measurements by himself and others to calculate the rate of heat-flow into the piston, in C.H.U. per min., and expressed this as a fraction of the total heat of combustion of the fuel per cycle. His results are summarized in table 20. Considering the very diverse

TABLE 20

Proportions of the total heat of combustion which passes to the piston, calculated from piston temperature measurements.

Authority	*Engine type and speed*	*Piston bore and stroke and material*	*B.H.P.*	*Heat to piston as a percentage of total heat of combustion*
Hopkinson . .	Gas engine 180 r.p.m.	C.I. 11·5 in. × 21 in.	15·6	3·6
Burstall . . .	Gas engine 170 r.p.m.	C.I. 16 in. × 24 in.	11·8	3·7
Coker . . .	Gas engine 200 r.p.m.	C.I. 7 in. × 15 in.	16·2	4·6
Gibson . . .	Petrol engine 1,800 r.p.m.	Aluminium 100 mm. × 140 mm.	13·9	3·5

types of engines covered by the table the uniformity of the calculated proportion of heat to the piston is satisfactory. The mean of the figures in this table is also in very close agreement with the figure to be arrived at from the analysis of heat losses given in art. 61 (i): an independent analysis, based purely on considerations of the probable subdivision of the total heat losses in an engine of compression ratio 5 : 1.

The figure one derives from that analysis, for the fraction of the total heat of combustion which finds its way to the piston, and thence to the cylinder walls, is 4 per cent., as compared with the mean of the figures in table 20 of 3·9 per cent.

Apart from numerical values for the heat quantities, the forms of equations (12) and (13) bring out several points of interest. They show, for example, that when the piston crown is tapered in thickness the temperature difference between the edge and the centre will be proportional only to the first power instead of to the square of the radius; and that the temperature difference will be inversely proportional to the conductivity of the material. As regards the thickness of the crown, it is probable that in high-speed engines where lightness is important the crown will not be made thicker than is essential for strength. It must be remembered that h, in equations (12) and (13), represents the difference between the heat gained and lost on the two sides of the piston, and that even from the cooling point of view, therefore, additional thickness might only have the effect of passing to the cylinder walls some of the heat which would otherwise have been carried off by oil-splash from the underside of the piston. It would not necessarily produce a cooler piston.

It is of interest that the temperature differences shown in fig. 35 for the cast-iron and aluminium pistons are very nearly in the inverse ratios of the conductivities, namely as 4 to 1, so that the ratio h/t would appear to have been the same for each piston. As mentioned earlier, however, it is probable that the rate of heat absorption by the cast-iron piston was affected by a carbon deposit as well as by oil-splash.

One further point may be mentioned. The rate of heat-flow into the piston and cylinder walls is, of course, periodic, fluctuating with the cylinder gas temperatures. There will therefore be a fluctuation of the surface temperature of the metal, which can be calculated in terms of the constants for the material, and of the amplitude and periodic time of the temperature fluctuation. Taking the values for e given in table 19, Gibson[11] estimated that in a petrol engine at 1,800 r.p.m. the temperature fluctuation of the surface of an aluminium piston would be $\pm 5{\cdot}5°$ C., and of a cast-iron piston $\pm 16°$ C. At high speeds, therefore, the amplitude of the temperature fluctuation is of no practical importance.

Art. 25. *The exhaust valve.*

Thanks to the skill of the metallurgist in developing steels to withstand high temperatures without scaling, and without serious loss of mechanical properties, it is possible for exhaust valves to survive

many hours of continuous operation at a bright red heat, representing temperatures of the order of 750–800° C. Although the valve itself may stand it, however, it is evident that the presence of valve-heads at such a temperature, comprising as they do some 20 per cent. of the walls of the combustion space (excluding the piston), must have a profound effect upon the successful operation of the cylinder.

Apart from the direct effect of the hot valve-heads in heating up the incoming charge, reducing the volumetric efficiency, and producing detonation, there is the indirect effect of the radiation from them which is an important element in the heating of the piston. During some experiments to be mentioned again in the next article, in which the two exhaust valves in an air-cooled cylinder were individually cooled and so kept down to about 300° C., it was found that because of the improved volumetric efficiency with the cooled valves an average increase of 2·5 per cent. in the maximum power was obtained. And, moreover, the maximum power obtained with the uncooled valves could be obtained with the cooled valves at a petrol consumption 8 per cent. lower than without them.

Because of its effect upon combustion, and upon the temperature, and hence the lubrication, of the piston, this question of exhaust valve temperatures is one of primary importance. One would anticipate (see art. 61 (i)) that the most severe conditions for the exhaust valve would be in an inefficient engine of low compression ratio, or when a high compression engine is working with a weak mixture or retarded ignition. In either case the exhaust gas temperature tends to be high, and as a consequence the valve temperature also.

During every exhaust stroke both the valve head and stem are subjected to the scouring action of the hot exhaust gases flowing at a high speed; and during the remainder of the cycle the heat then received must be got rid of by radiation, and by conduction to the valve seat and valve guide. When the valve seat is an inserted ring of hard metal (as illustrated in figs. 2 and 4), then perfect metal to metal contact between it and the aluminium cylinder head is evidently essential, and any loosening of the seat while in service would be the preliminary to rapid overheating and burning out of the valve. The most usual type of failure, commonly spoken of as the 'burning out' of an exhaust valve, may be expected to start by a failure of the valve to make a gas-tight joint with the seat. Due either to distortion or to the nipping of a piece of carbon or scale between the valve and its seat, a minute passage is left through which gas can escape under high pressure and at the highest explosion temperatures. As soon as such a passage has been established its enlargement by melting of the edge of the valve is likely to follow rapidly.

Measurement of the mean temperature of the head of an exhaust valve in a high-speed petrol engine is a matter of some difficulty. Direct measurements by means of a thermocouple have been made, but at high speeds the technique becomes extremely troublesome. Such measurements were employed by Gibson[12] to check the readings of an optical pyrometer which was focused, through the open exhaust port, upon a point in the upper surface of the valve head midway between the periphery and the centre. Under normal conditions

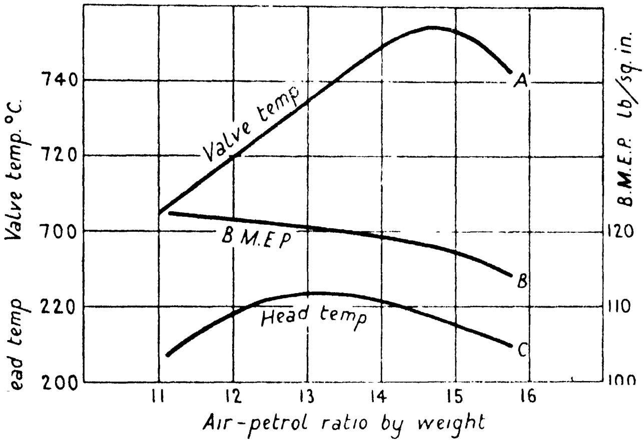

FIG. 39. Valve and head temperatures of an air-cooled cylinder, 100 mm. × 140 mm. bore and stroke. C.R. 5·5. Speed 1,800 r.p.m. Cooling air at 62 m.p.h. and 19° C. Ignition adjusted for maximum power without detonation.

this is the hottest point, for the periphery is cooled by contact with the seat, and the stem by the transfer of heat to the valve guides. Comparative readings of the pyrometer and thermocouple showed that with care the pyrometer was capable of giving exhaust valve temperatures with an accuracy of ± 10° C.

The dependence of the exhaust valve temperature upon the air-petrol ratio is illustrated by the curve *A* in fig. 39, reproduced from Gibson's observations. The simultaneous variations of the B.M.E.P., and of the cylinder head temperature near the valve seat, are shown by the curves *B* and *C*.

It will be noticed that the maximum valve temperature was obtained at an air-petrol ratio of 14·7 as compared with 13·1 for the maximum cylinder head temperature. The latter is less sensitive to changes in the air-petrol ratio, and the changes of temperature, such as they are, reflect the maximum gas temperatures before expansion; while the valve temperatures reflect rather the temperatures of the escaping gases *after* expansion. Hence follow the higher temperatures

with the weaker, and therefore slower burning, mixtures. With mixtures weaker than 15 : 1 the reduction of the fuel, and consequently of the gas temperatures throughout the cycle, more than counterbalances the effect of the slow burning. In a further set of experiments it was found that the exhaust gas temperature, measured above 15° C., was 19 per cent., or about 200° C., higher at an air-petrol ratio of 15 than at 13·5. The former was the ratio which gave a maximum thermal efficiency on the brake. The latter was the weakest mixture for maximum power. Under normal conditions it was found that the mean valve temperature in the air-cooled cylinder was 15° C. higher at 2,000 than at 1,800 r.p.m., so that it is clear the influence of the air-fuel ratio upon the valve temperature is likely always to be more important than that of speed, within the usual speed-range of aero-engines.

It might be supposed that in a water-cooled engine the problem of cooling the exhaust valve would be less difficult. Comparative experiments upon aero-engine cylinders are not available, and it may be that in the cast-iron cylinder used by Gibson the cooling of the valve seats was less effective than it would be in an aluminium head. In Gibson's experiments, in spite of the water outlet temperature being maintained at about 60° C., the temperature of the head round the edge of the valve seat was as shown in fig. 40. The head temperatures there shown for different air-fuel ratios are the mean values of four thermocouple readings close to the valve seat.

The curves in figs. 39 and 40 show that at the weakest mixture for maximum power, 13·5 : 1, the valve temperature was only about 12° C. cooler in the water-cooled cylinder. At other air-fuel ratios the conditions are not properly comparable, owing to the constancy of the ignition advance adopted with the water-cooled cylinder. The effect of this is reflected in fig. 40, in the rapid fall of power, accompanied by a maintenance of high valve and cylinder-head temperatures even with the weakest mixtures, owing to there having been no extra ignition advance to counterbalance the effect of the slower burning.

For the same reason the maximum valve temperature in the water-cooled cylinder was greater than in the air-cooled, 762° C. as compared with 755° C., and it was reached at an air-petrol ratio of 15·8, as against 14·8 when the ignition advance was adjusted to the optimum for each mixture.

Ignition advance, as would be expected, has a very pronounced influence upon the exhaust valve temperature. Too great an advance will be liable to cause detonation and consequent overheating on that account, but it may be given as a general conclusion that for all com-

pression ratios the lowest valve temperatures, the highest powers, and the lowest petrol consumptions are obtained with the spark as fully advanced as is possible without causing detonation. This optimum ignition advance must depend upon the speed and the compression ratio, but for the cylinder we are now concerned with, running at 1,800 r.p.m., the magnitude of the effect of ignition advance upon the head and valve temperatures, and upon the B.H.P., were as shown in fig. 41.

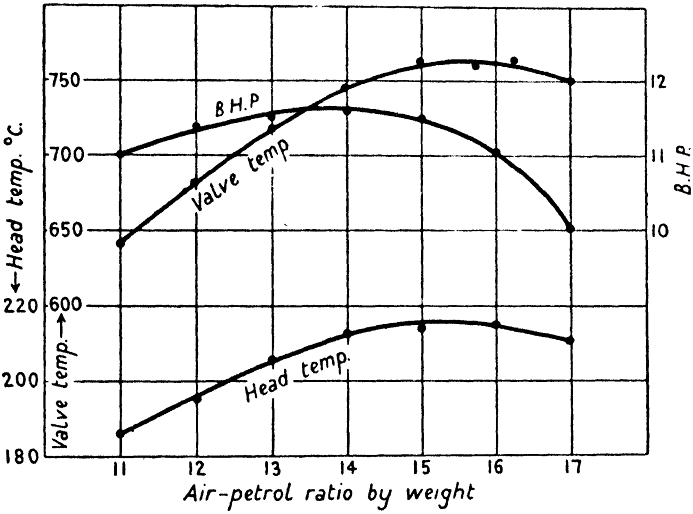

FIG. 40. Exhaust valve and cylinder-head temperatures of a water-cooled cylinder, $3\frac{1}{2}$ in. × $5\frac{1}{4}$ in. bore and stroke. C.R. 4·77. Speed 1,800 r.p.m. Jacket water outlet approx. 60° C. Ignition timing constant at 45 deg. advance.

So long as the advance is not so great as to cause detonation, its effect upon the head temperature is slight; but if detonation occurs there would be a substantial rise above the figure of 198° C. shown in fig. 41. A decrease of the advance below the optimum value, while increasing the exhaust gas temperature, will at the same time diminish the maximum explosion temperature and pressure. The exhaust valve suffers the full effect of the hotter escaping gas and is very sensitive to its temperature, but the much slighter effect of this upon the cylinder head is more than compensated by the less heat received during the earlier part of the cycle. The result is the slight fall in the head temperature shown in fig. 41, as the ignition advance is reduced.

Since ignition advance has so marked an effect, it is only to be expected that the placing of the sparking plug in relation to the valves should also affect their temperatures. It is unsafe to generalize widely in the matter, for the influence of the plug position must

depend to some extent upon the prevailing flow of gas in the cylinder during combustion; and this in its turn upon the size and arrangement of the inlet valves. Gibson found, for example, that when using two alternative plug positions which were absolutely symmetrical with respect to the valves, both the power and the exhaust valve temperature were quite substantially affected according to whether the plug was in a horizontal position or was inclined at 35° to the cylinder axis.

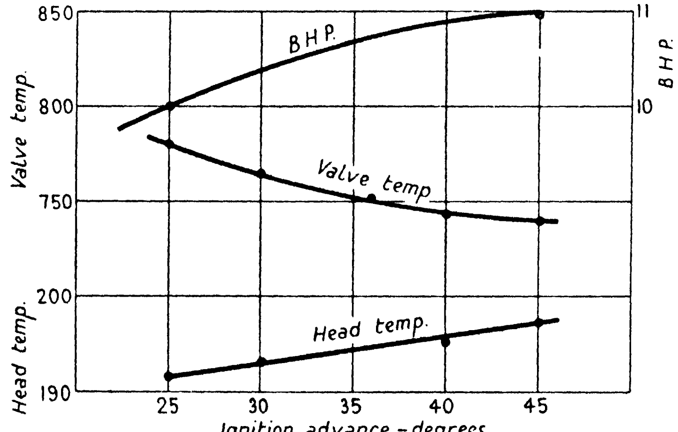

FIG. 41. The influence of ignition advance upon exhaust valve and cylinder-head temperatures. Water-cooled cylinder. C.R. 4·26. Speed 1,800 r.p.m. Weakest mixture for maximum power.

Apart from such details, which can only be settled by experiment with any particular design, it is inadvisable to fit a sparking plug either directly adjacent to, or directly opposite, an exhaust valve. During the progress of combustion the gases in the vicinity of the plug, already at a very high temperature due to combustion, have their temperature further raised by compression. As a result the gas temperatures near the plug are likely to be higher than in other parts of the cylinder, and if the exhaust valve is on the same side its temperature and that of its seat are likely to be higher than if it is remote from the plug.

On the other hand, if the plug is opposite an exhaust valve, so that the gases already compressed in the neighbourhood of the hot valve are the last to burn, then this is a state of things more prone than any other to set up detonation. Of the two alternatives the latter is likely to be the more fatal to satisfactory operation, so that if it is impossible to avoid placing a plug either opposite or adjacent to an exhaust valve this last alternative is the least objectionable.

Finally, as regards the effect of a change of the jacket-water temperature, this was found by Gibson to be much the same over the whole range of speeds and compression ratios. Typical results, obtained upon an automobile type of cylinder, are shown in fig. 42. The highest and lowest curves of head temperature each represent the average of a group of thermocouples round each valve; while

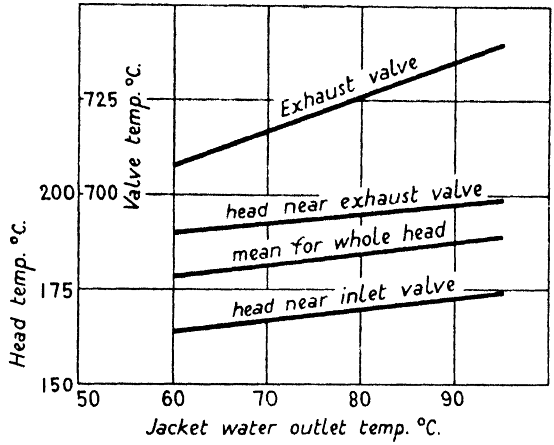

FIG. 42. Exhaust valve and head temperatures in a water-cooled cylinder, as affected by jacket water temperature. C.R. 4·77. Speed 1,800 r.p.m.

the intermediate curve shows the mean of all the head temperature readings. Over the whole range of observations represented in this figure the B.H.P. and rate of fuel consumption of the cylinder did not vary by more than 1 per cent., the reduction of bearing and piston friction at the higher temperatures apparently compensating almost exactly for the reduced volumetric efficiency.

The curves of fig. 42 are interesting, as showing the small effect upon the head temperature of a change in that of the jacket water. For an increase of 35° C. in the water temperature it will be seen that the rise in the mean head temperature was only 11° C.; while that in the valve was 32°, almost the same as the rise in the water.

The temperature difference between the cylinder head and the water became less as the temperatures of both increased, but at the highest water temperature the mean temperature for the whole of the head was still 94° C. above the water. These are observations upon a cast-iron cylinder, but further tests upon a typical aero-engine, summarized on p. 212, show that the cylinder-head temperatures were as much as 60°–70° C. above the water outlet temperature.

It is obvious that the 32° C. reduction in the valve temperature between the extremes of the water temperature in fig. 42 can have

been due to a small extent only to a cooler valve seat; the greater part was probably due to a cooler valve guide and a more rapid loss of heat from the stem. From measurements of the heat passing to the jackets, and that remaining in the exhaust gas, it was shown that the rise of 35° C. in the water was accompanied by a diminution of 8 per cent. in the amount of heat carried off by it, and by a rise of the mean exhaust gas temperature from 998° to 1,073° C. The rise of the valve temperature which accompanied the hotter jackets was therefore due to the combined effects of a hotter valve guide, hotter exhaust gas, and to a slight extent also to a hotter valve seat.

Art. 26. *Sleeve-valves.*

In some experiments by Gibson already referred to, of which the results are summarized in table 21, the single exhaust valve on two different air-cooled cylinders was cooled artificially by including a fluid within the hollow stem and designing a tiny radiator at the end of it. It will be seen that it was found possible to reduce the temperature of the hottest point of the valve head in this way by about 450° C. and that, with no other change, this resulted in an increase of the maximum power obtainable of about 4 per cent. in the first series of tests quoted in table 21.

TABLE 21

The effect upon the maximum power output of an air-cooled cylinder of using an artificially cooled exhaust valve.

Cylinder and valve size	*r.p.m.*	*Compression ratio*	*Maximum B.M.E.P.*	*Petrol pts. per B.H.P. hour*	*Exhaust valve temp.*	*Valve*
100 mm. × 140 mm.	1,800	4·7	126·0	0·59	260° C.	Cooled
Valve 42 mm. diam.			121·0	0·59	700° C.	Uncooled
	1,800	5·5	133·5	0·59	300° C.	Cooled
			127·5	0·615	750° C.	Uncooled
5½ in. × 6¼ in.						
Valve 2·3 in. diam.	1,650	5·0	117·0	0·60	400° C.	Cooled
			113·4	0·62	760° C.	Uncooled

The mean temperature of the cylinder head was lowered by 26° C. and the hottest point of the piston by 25° C. as a result of the reduced conduction and radiation from the hot valve.

The results of a large number of tests with different cylinders showed that on an average an increase of 2·5 per cent. in the maximum power could be obtained by cooling the valve; and that the maximum power obtainable with an uncooled valve could be maintained, when keeping the valve cool, with a petrol-air mixture so

much weaker as to reduce the rate of petrol consumption per B.H.P. hour by 8 per cent.

When the breathing arrangements of a cylinder are provided, not through poppet-valves in the head, but through ports cut in a cylindrical sleeve (see fig. 43) which surrounds the piston and moves in relation to it, all the advantages of a cooled exhaust valve are achieved. The thin steel sleeve-valve* is in contact over nearly the

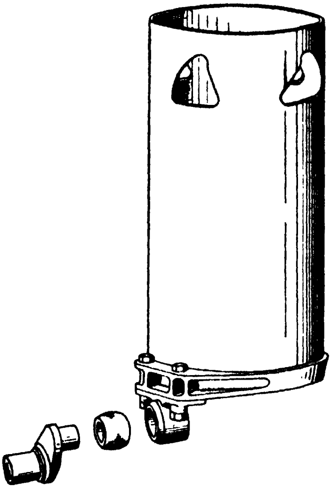

FIG. 43. Sleeve-valve and driving mechanism.

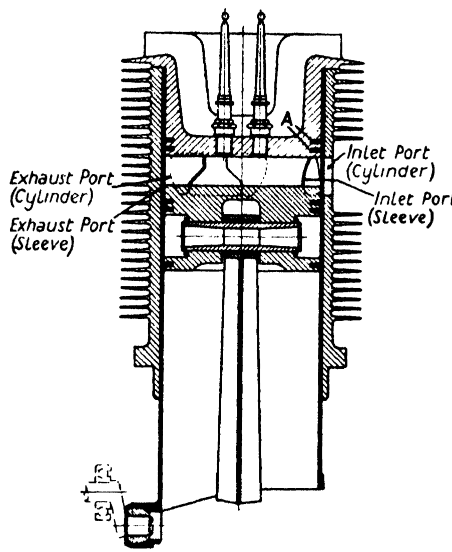

FIG. 44. Cross-section of air-cooled sleeve-valve cylinder.

whole of its outer surface with the cooled cylinder barrel, and no part of the inner surfaces of the cylinder should be above about 200° C.

Unlike the plain reciprocating motion of the sleeves in a double sleeve-valve engine, the single sleeve is given a reciprocating and at the same time a rotating motion, by means of a mechanism operated by a shaft rotating at half engine speed. The method of driving the sleeve by a small crank through a ball-and-socket joint will be understood from figs. 43 and 44. The motion is such that any point on the sleeve describes approximately an ellipse of which the major and minor axes are in practice in the ratio of about 3 : 2. This

* In the Daimler and other successful road vehicle engines two concentric sleeves have been employed, but the power output required from an aero-engine cylinder makes it doubtful whether the double sleeve system could ever be successful in such an engine. The difficulty of cooling the piston and controlling the oil supply would be too great. Single sleeve-valves only, of the Burt type, will therefore be considered here. For a fuller discussion the reader is referred to a paper by Ensor.[13]

enables the sleeve to be designed so as to uncover alternately exhaust and inlet ports cut in the cylinder walls.

In consequence of the absence of any hot exhaust valves it is possible to employ a compression ratio, without detonation, which is from a half to a whole ratio higher in a sleeve- than in a poppet-valve cylinder of the same size and with the same fuel. And furthermore, even when detonation does occur, its onset is less sudden and its effects are less severe, since it does not readily develop into pre-ignition. The piston, it is true, has to get rid of its heat through the sleeve and across two pairs of lubricated surfaces instead of directly across one; but it must be remembered that the cooling of the exhaust valve in Gibson's experiments lowered the maximum piston temperature by 25° C. owing to the absence of radiation from the hot valve, and the piston in a sleeve-valve cylinder therefore starts with at least this advantage.

Since the valve ports are no longer crowded within the surface of the cylinder head, the design permits of large port openings in relation to the swept volume of the piston, especially in the smaller sizes of cylinder, and high volumetric efficiencies are therefore obtainable at high speeds. Reference has already been made in Chapter III to the small sleeve-valve cylinder from which Ricardo obtained a B.M.E.P. of 160 lb. per sq. in. at 3,000 r.p.m. when normally aspirated, and 140 lb. per sq. in. at 5,000 r.p.m., the C.R. being 6·8 : 1 and the fuel a straight-run aviation spirit with no added benzol or anti-knock dope.

A sleeve-valve cylinder gains also in the freedom of the designer over the shape of his combustion space, when this has no longer to accommodate 2 and often 4 valves, as well as 2 sparking plugs. In a sleeve-valve cylinder the combustion space can be made any shape which is best for the avoidance of detonation, provided only that in an aero-engine a suitable location for 2 sparking plugs can be found. Apart from this proviso a combustion space in the shape of a truncated cone, with one plug at the apex, seems to have many advantages from the point of view of detonation.

The single sleeve-valve has been slow to make its way as a practical success in aero-engine design, not so much from any fundamental disadvantage as because its successful introduction brings with it a set of new problems in detailed design which can only be solved by experience, gained at the cost of discouraging failures by the way. There is a vast amount of accumulated experience behind the successful poppet-valve of to-day, and in face of the appalling expense of a failure in any radically new design of engine the natural course is to go on improving the poppet-valve, step by step,

rather than to launch out upon a new line of development beset by pitfalls.

The main difficulties of the sleeve-valve cylinder are centred in the problems of gas tightness and oil control; and these in their turn upon the maintenance of the correct working clearances between the piston, sleeve, and cylinder barrel. In a light engine the sleeve must be of steel, about $\frac{1}{10}$ to $\frac{1}{8}$ in. thick, while the piston and cylinder are of aluminium alloy. Unless a specially high expansion steel can be found for the sleeve there will be obvious difficulties about the maintenance of the correct clearances, in view of the coefficients of expansion of aluminium and iron, the one about double the other. Research has in fact provided a nickel-chrome-iron alloy with a coefficient of $18–20 \times 10^{-6}$ per deg. C., which is only a little less than that of the light alloy used for the cylinder, namely, $22{\cdot}5 \times 10^{-6}$.

In order to provide a gas-tight joint at the top of the cylinder it is necessary to fit 'junk rings' on the cylinder head as shown at *A* in fig. 44, and these provide a potential source of trouble in the form of gummed lubricating oil which sticks up the rings, for they are located in that portion of the head which it is most difficult to cool. This is more especially so in an air-cooled cylinder such as that illustrated, for it is almost impossible to provide really effective cooling in the deeply recessed end of the cylinder, and reliance has to be placed upon conduction through the sleeve to the outer cylinder wall for getting rid of the heat from the neighbourhood of the junk rings.

The lubrication of the sleeves has not been found a serious difficulty, nor is the power required for driving them such as to give an engine a low mechanical efficiency. The combined rotary and reciprocating motion maintained by the sleeve is particularly favourable to a good distribution of the oil film, and the wear, either of the sleeve or of the outer cylinder, has been found to be negligibly small even after thousands of hours of running. Owing to the large area of lubricated surface on the outer side of the sleeve the forces necessary to operate the sleeves may be large just at the start, while the oil is viscous, and this would be a serious problem if an engine had been allowed to get very cold before starting up.

There are certain mechanical and manufacturing points where the balance is strongly in favour of the sleeve-valve, and these, combined with its other advantages, may well enable it to win an important place ultimately in aero-engine design. As crankshaft speeds are increased, for example, the cam-and-spring type of movement of the poppet-valve is a constant source of anxiety. To provide the necessary acceleration more and more powerful springs are called for, and higher working stresses in the material. Every valve needs at least

2 springs, making the formidable total for a 12-cylinder engine of 96 springs, with proportionate danger of failure through slight faults of manufacture. For the sleeve-valve, on the other hand, with its smooth and continuous motion, increases of speed are immaterial.

From the point of view of manufacture there is the advantage of a big reduction in the number of separate parts per cylinder, and of the comparatively simple nature of the machining processes involved. It would, however, be inappropriate to stress this point too much, in view of the very limited manufacturing experience with sleeve-valve engines. Suffice it to say that, such as it is, it has indicated the possibility of a substantial reduction in the cost of manufacture per h.p. as compared with present-day engines, which involve all the very expensive hand work necessary for the shaping and fitting of poppet-valves suitable for high speeds and a high B.M.E.P. It is not sufficient to use special steels, and seating materials such as stellite.* Complex designs of liquid-cooled valves are now widely used to reduce valve-head temperatures. The stem is made hollow, and after filling to about 60 per cent. of the volume with metallic sodium, it is closed at the end by a steel plug, commonly swaged in. When the valve is working the sodium melts and carries heat from the head to the stem, and thence to the valve-guides, for direct dissipation of the heat by a small radiator on the stem, as used by Gibson, would hardly be practicable in service.

Apart from the elaborate and costly design, these special cooling devices can hardly fail to make the poppet-valve heavier, and high-speed operation more difficult, so that, by offering at the same time a cool valve and an absence of large accelerations, the sleeve-valve offers a big return for its successful development.

Finally, on the question of relative fuel economy, the sleeve-valve should show to advantage for three distinct reasons. It was mentioned at the beginning of this article that with a cooled exhaust valve Gibson was able to maintain the same maximum power in his experimental cylinder at a fuel consumption per B.H.P. lower by 8 per cent. than with the uncooled valve. This was because it was not necessary to employ a rich petrol-air mixture to help to cool the cylinder and suppress detonation. Besides the possibility of a higher compression ratio without detonation, which should afford an increase of the intrinsic thermal efficiency of the cylinder of some 5 per cent., there is the third advantage that all danger of burnt exhaust valves, when operating upon weak and therefore economical mixtures, disappears; and with it all hesitation about weakening the mixture, when cruising, to obtain maximum economy.

* For further information on this and on sodium filling see the paper by Banks.[69]

V

LUBRICATION AND LUBRICANTS

ART. 27. *An outline of the problem.*

At every one of the points in an engine where metal surfaces are in contact, and in relative motion, heat is produced by friction. The problem of lubrication is that of reducing the rate of production of this heat at every point to a minimum, and to something less than the rate at which it can be removed. Between some surfaces in motion past one another it is possible to maintain an unbroken oil film, so that the metal surfaces are never actually in contact. When this is so the condition is described as one of 'fluid' or 'complete' lubrication. The frictional force which opposes the relative motion of the surfaces then depends primarily upon the viscosity of the oil film, and could be calculated in terms of the properties of the oil, if the thickness and temperature of the film at every point were known. The tangential force is almost independent of the normal force between the surfaces; and is proportional to the speed of relative motion and to the area of the surface.

When conditions are not such as to maintain an oil film of finite thickness between the metal surfaces, the state of affairs is one which is described as 'boundary' lubrication. The frictional force then depends on the condition of the lubricated metal surfaces, which are intermittently in contact with one another, at any rate in the sense that they are not electrically insulated. The characteristics of this state of boundary lubrication will be considered in detail in a later article, and it will be sufficient here to point out that the observed friction, while always lower than that for unlubricated surfaces, is far above that when fluid friction prevails. As compared with coefficients of the order of 0·01 with fluid friction, but varying widely according to the conditions, those with boundary friction would be in the range 0·1–0·3. The frictional force under conditions of boundary lubrication is found to depend in a very complex manner upon the condition of the oil and of the surfaces, and even upon the chemical nature of the oil and its reaction with the surface metal.

It should be remembered when thinking of a lubricated bearing that no surface in practice is either perfectly flat or perfectly cylindrical. As we approach the condition of boundary lubrication we are concerned with differences of level in the surface amounting to no more than a few diameters of an oil molecule. Considered on this scale, even a ground steel shaft, and, much more, the white metal lining of

a bearing, must be pictured as having an undulating surface of hills and valleys. As the surfaces slide over one another, points of contact under conditions of boundary lubrication will alternate with areas where a finite thickness of oil film separates the surfaces.

The relation between the normal and tangential forces for surfaces in a state of true boundary lubrication can be measured in a laboratory in a carefully arranged experiment, but the information obtained would only be applicable to a very small fraction of the rubbing surfaces in an internal combustion engine. It has been supposed that the state of affairs between a piston ring and the cylinder wall must be one of boundary lubrication, but there is indirect evidence that even here the friction has the magnitude and nature of the fluid type, except possibly at the ends of the stroke. Everywhere else, except perhaps between the teeth of gears and occasional high spots in a bearing, an oil film of varying thickness is maintained and the frictional forces are those characteristic of fluid motion.

The lubrication of pistons in their reciprocating motion will have to be considered separately from that of bearings and gears, by reason of the severe conditions to which the oil film on the cylinder walls is subjected. During every up-stroke a fresh supply of oil is carried up by the piston, and the film of this which is left behind is swept for half the time by gases at temperatures between 1,000° and 2,000° C. This cannot be without effect on the surfaces of the oil film, either in the way of 'cracking' the oil molecules to form unsaturated and gum-forming hydrocarbons of smaller molecular weight, or of partial oxidation when free oxygen is present in the cylinder gases. After the working stroke the altered surface film is overrun by the piston again, and some of the products of the cracking and oxidation processes find their way into the spaces around the rings near the top of the piston. In a high duty engine the piston itself may reach temperatures ranging from 250° C. at the centre to 200° C. at the periphery under normal working conditions, and the association of these temperatures in the metal with an accumulation of cracked and oxidized oil off the cylinder surface prepares the way for the sticking of piston rings. This is brought about by the fatal metamorphosis of the oil into the very antithesis of a lubricant, namely into a sticky, glue-like substance, insoluble in the original oil and, in its final stages, insoluble in almost anything.

Between the cylinder wall and the outer surface of the piston rings the condition of boundary lubrication may prevail to some extent, more especially during the early part of the expansion stroke when the high gas pressure, getting behind the rings, may largely increase

the normal pressure between them and the cylinder walls. To a lesser degree it may happen also at the end of the stroke, when the piston velocity again drops to zero. During the central part of the stroke the high piston speed enables the ring to ride-up on an oil film of appreciable thickness, and in these conditions the frictional force is largely unaffected by the surface condition of the metal and of the oil. Between the piston skirt and the cylinder wall the conditions may approach those of boundary lubrication over a small part of the circumference, on the side which is subject to the thrust of the connecting-rod; but round the rest of the circumference they will everywhere be those giving viscous friction.

That an engine as a whole exhibits the phenomena of fluid rather than solid friction has been proved by experiments upon the rate of deceleration of a freely running engine when the power is cut off, for the power absorbed has been shown to be proportional nearly to the square of the speed. There is also the point that the friction losses observed when motoring engines while they are still hot, after running under power, are found to correspond closely to the viscosity of the oil used, and to show up none of the large differences between lubricants which can be demonstrated under conditions of boundary friction. For example an engine, lubricated with a fatty oil which gave a coefficient of 0·03 under nominal boundary conditions,[14] showed a higher friction loss than when lubricated with a mineral oil of which the boundary friction coefficient was no less than three times that of the fatty oil. Furthermore, if a rough estimate is made of the mechanical efficiency and the heat generated in the bearings of an engine, using boundary friction coefficients, it will be found that the mechanical efficiency is quite impossibly low, and that the heat generated in the bearings must inevitably lead to seizure. We may conclude, therefore, quite definitely, that by far the major portion of the friction loss in an engine is by fluid friction, with boundary conditions occurring only from time to time between the 'high points' of two bearing surfaces and possibly between the cylinder and piston rings near the two ends of the stroke. Although the friction loss during normal running depends upon the viscosity of the lubricant, the safety of the engine from seizure, on the other hand, will by no means be independent of the characteristics of its lubricant under boundary conditions.

It is clear that a primary requirement in a good lubricant where the cylinder and piston surfaces are concerned must be chemical stability at high temperatures. At the bearings and gears, on the other hand, the temperatures need never be so high as to approach the danger point for chemical stability, and the first essential is that

the oil should be able to maintain lubrication under very high local concentrations of load; in other words, that the coefficient of friction should be low under conditions of boundary lubrication.

It happens that the oils of greatest chemical stability at the piston temperature are not those which serve best in boundary lubrication. The ideal, therefore, would be to have a separate oil supply for the cylinders, and for the bearings and gears. Such an ideal, of supplying two different oils, however, is impracticable in aero-engines, and development has taken the line of employing mineral oils for their greater chemical stability, and of meeting any deficiency they may show under boundary lubrication by more perfect mechanical design, to minimize local concentrations of stress. By the more perfect forming of gear teeth, for example, excessive local pressures can be avoided and gears can be made to function satisfactorily with a mineral oil, which formerly were liable to abrasion of the tooth surfaces. Under these severe conditions the vegetable castor oil shows a remarkable superiority over other lubricants which has nothing to do with its high viscosity. The somewhat obscure property of 'oiliness' possessed by castor and other 'fatty' oils to an exceptional degree, will be further discussed in art. 29 on boundary lubrication. It depends upon the power of certain types of oil molecules to arrange themselves in a definite orderly manner when forming primary films on a metal surface. These films may in the limit be not more than one molecule thick, and the molecules which compose them are held to the surface of the solid by powerful bonds essentially similar to the bonds of a chemical union. The film is said to be 'adsorbed' on the surface of the metal, and it adheres so powerfully as to be removable only with the greatest difficulty.

This difficulty in the complete removal of the primary film from a metal surface is a feature of all lubricants which is only especially noticeable in the case of castor and other fatty oils. The facts can easily be demonstrated by the behaviour of an experimental bearing when lubricated by a succession of different oils. The behaviour, both as regards coefficient of friction and liability to seizure, is always found to depend not only upon the particular oil in use at the moment, but upon the previous history of the bearing. Until this was realized it naturally led to a great number of apparent inconsistencies among lubrication experiments, however carefully conducted. It has been shown by Deeley, Stanton, Hardy, and others that in some cases nothing short of the regrinding of a metal surface will remove the last traces of a lubricant. Washing with solvents is by no means effective.

The beneficial effect of castor oil upon lubricated surfaces, and its

persistence after removal of the oil supply, has led to the practice of 'running-in' new aero-engines for several hours upon castor oil, by which time the surfaces have become thoroughly impregnated, and thereafter turning over to mineral oil for the rest of the engine's life. The unsatisfactory qualities of the castor as a piston lubricant, by reason of its propensity to oxidation and gum formation, would not become evident until after a hundred hours or more of running, so that its use for a few hours with new engines allows some of its admittedly superior qualities for bearings and gears to be retained without subsequent danger from the sticking of piston rings.

On the question of the part played by the viscosity of an oil, and of its importance, there are many opinions but there is little certainty. It is customary to specify a minimum viscosity for an oil at a certain temperature, and when engines have been designed to work with oil of a certain viscosity it is not safe to change that viscosity without simultaneous attention to bearing clearances, piston-ring design, and the working oil pressure which is maintained by the relief valve. A simple lowering of the viscosity of the oil used would allow of a more rapid flow through the shaft bearings, and better cooling; but it would mean also that more oil per revolution would be thrown on to the cylinder walls and this would lead to an excessive rate of oil consumption unless provision against it were made by redesign of the bearings and of the piston scraper rings.

How far a high viscosity is a safeguard against lubrication failure, either at the pistons or gears, is a question to which there is no simple answer. When once the oil film has been squeezed out, and boundary lubrication has set in, with the surfaces in contact, there is no doubt at all that the viscosity of the lubricant can play no further part in preventing seizure. But seizure under these conditions will depend not only on the lubricant, but also, if the load is sufficiently severe, upon how long the condition of boundary lubrication persists. The more viscous the lubricant the longer will it take to be squeezed out from between two surfaces, and, remembering the undulating nature of every bearing surface considered in relation to films of molecular dimensions, it is quite clear that an oil of low viscosity will in general allow longer periods of metallic contact between the high points of a bearing, for a given load per unit area. On the other hand, if the less viscous oil should give a lower coefficient of friction in boundary lubrication, the rate of heat production would be less, and seizure less imminent.

It will be seen, therefore, that although viscosity plays no part once the metal surfaces are in contact, it may yet be a safeguard against seizure through its delaying the occurrence of that contact.

How far viscosity is an effective factor in maintaining the oil film depends very much on the type of surface considered. Between a cylindrical journal and its bearing it is probably of the first importance, for the eccentricity of the journal allows, indeed it induces, a constant supply of fresh oil at the point of maximum load concentration.[15] As between a piston ring and the cylinder wall, however, it is doubtful whether viscosity can help so much to prevent the metallic contact, although here again, if one is right in thinking that boundary lubrication only sets in at the two ends of the stroke, it can be argued that it will set in sooner with a less viscous oil.

It may be, but it does not necessarily follow, that an oil of low viscosity is also more volatile than a more viscous one. If that is so, then it must increase the danger of piston seizure with the less viscous oil, for towards the end of the exhaust stroke in a 4-cycle engine the piston and rings are dependent for lubrication upon what is left upon the cylinder walls after exposure for nearly a whole revolution of the shaft to temperatures of 1,000°–2,000° C. In these circumstances the danger of a dry wall must be directly dependent on the volatility of the oil.

ART. 28. *The chemical nature of lubricants.*

The composition and chemistry of lubricants is so complex that no attempt will be made to deal thoroughly with it here. If the characteristic behaviour of the different classes of oils in an engine is to be understood, however, it is necessary first to grasp the essential features of their chemical constitution. For further information, reference must be made to the standard books[16] on the subject.

The two main classes of oils with which we are concerned are the mineral oils, and the 'fatty', or 'fixed', oils of which by far the most important is castor oil. This is of vegetable origin, as also is rape oil, perhaps the next in importance, the former being obtained from the castor oil bean, and the latter from rape seed, by crushing under high pressure. Fatty oils derived from fish and other animals are roughly similar in chemical composition, and need not be separately dealt with. The oils of animal and vegetable origin have been called 'fixed' oils because, unlike the mineral oils, it is not possible to distil them without decomposition.

The principle dividing line from the chemical point of view between the mineral and the fatty oils lies in the fact that the molecules of the latter contain oxygen while those of the former do not. The mineral oils are all composed of hydrocarbon molecules, of extreme complexity and variety, but all of the general type

$$C_nH_m.$$

When it is recognized that among the lubricating oils n may vary, perhaps, between 15 and 25, and m between 30 and 50; and that for every pair of values of m and n there may be vast numbers of isomeric molecules, containing the same total number of atoms but differently arranged, it is not surprising that an exact analysis of most oils is quite beyond the skill of the chemist. From partial analyses of oils from many different sources it has been concluded that the most satisfactory mineral lubricants contain only a small proportion of the saturated hydrocarbons of the open-chain paraffin series, of the general composition

$$C_nH_{2n+2},$$

and that they consist chiefly of types with less hydrogen in proportion to the carbon, of general formulae varying from

$$C_nH_{2n} \quad \text{to} \quad C_nH_{2n-8}.$$

The chemical behaviour of the oils leads, further, to the general conclusion that from 20 to 40 per cent. of most lubricating oils consists of unsaturated compounds, possibly of the open-chain type but more probably naphthenic, i.e. with molecules of a ring or a polynuclear structure. The rest of the oil would consist of saturated compounds, mainly of naphthenic,* and to some extent of aromatic* types.

In the absence of any exact knowledge of their composition, the mineral oils of commerce are classified according to the types of crude oil from which they come and the method of manufacture. The first stage in the treatment of the crude is the removal of the 'distillates' by heating under a fairly high vacuum (so as to keep down the necessary temperature of distillation) and their subsequent chemical treatment for the removal of resinous and asphaltic substances, and of any products of cracking during the distillation. To minimize these latter, superheated steam is usually blown through the boiling crude to carry off the hydrocarbon vapours rapidly from the still to the condenser, and so to minimize the time available for cracking to take place. Reduction of the temperature by high vacuum distillation also helps to reduce the amount of cracking which occurs.

The residues left behind in the still after removal of the distillates are the source, after refining, of the types of oil known as 'bright stock' and 'cylinder stock'. They constitute the least volatile types of mineral oils, and have been considered by some experts to be an essential ingredient of oils for the cylinder lubrication of internal combustion engines. This question will be further discussed in art. 33.

* See art. 16 (i).

It is stated above that the mineral oils are all complex hydrocarbons of the general formula C_nH_m, and to a very large extent this is true; but all types of oil are ready to absorb oxygen to some extent as the temperature is raised, and there is no doubt that lubricating oils contain at all times small amounts of compounds which include either oxygen or sulphur, or both. It is these oxy- and sulphur compounds which are the chief centres of gum formation while mineral oils are in use.

The fatty oils differ from the mineral in their containing oxygen as an essential constituent and in being to a far greater extent single chemical compounds. Their molecules are no less large and complex, but castor oil, for example, consists almost entirely of a definite chemical compound, in the molecule of which the number and grouping of the carbon, hydrogen, and oxygen atoms can be represented as follows,

$$C_3H_5[C_{17}H_{32}OH.CO.O]_3,$$

or, less explicitly, by $C_{57}H_{104}O_9$.

This is one of the class of compounds known as 'esters' which are formed by the combination of one of the alcohol group of compounds with a 'fatty acid'. The alcohol radicle occurring in the vegetable oils and in most of the animal oils is the trivalent group C_3H_5, which forms the radicle in the familiar compound glycerine of which the formula is

$$C_3H_5(OH)_3.$$

By replacement of each of the (OH) groups in the glycerine by the radicle of a fatty acid, represented for the moment by R, there are formed the series of esters known as glycerides, of general formula

$$C_3H_5R_3.$$

The particular fatty acid in the case of castor oil, known as ricinoleic acid, has a formula

$$C_{17}H_{32}(OH)CO.OH$$

and the composition of the glyceride which is the main constituent of castor oil is, therefore,

$$C_3H_5[C_{17}H_{32}OH.CO.O]_3$$

in which the complex radicle of the fatty acid within square brackets takes the place of the R in the general formula given above.

Castor oil contains a small proportion of the glycerides of other fatty acids, but this will only mean that the part of the molecule shown above in square brackets will differ slightly in composition.

All the vegetable oils, such as castor, rape, linseed, etc., are

generally similar in composition, but they differ in one important respect, of particular interest in connexion with their possible use in engines. Some of them, among which linseed is typical, belong to the class of 'drying oils' which, when left exposed to the air at ordinary temperatures, begin to react with the atmospheric oxygen to form gums and varnishes. Castor oil is obviously not one of this class, nor any of the others, such as rape and olive oil, which are used as lubricants, but their close cousinship to the drying oils is highly significant in the light of the known danger from gummed-up piston rings when internal combustion engines are lubricated with castor oil. Although the mineral oils may be oxidized and form gummy substances at high temperatures, their greater stability against oxidation is reflected in their complete inertness towards atmospheric oxygen.

The fatty oils are often described as 'saponifiable' on account of their reactions with soda and potash. When treated with caustic soda they form soaps, which are soluble in water; and this gives a ready way of detecting and separating a fatty oil from a mixture with a mineral oil, the latter being entirely unaffected by the treatment with the alkali.

One further point of general importance must be mentioned—that is, the ability of some oils to develop an acid reaction. Among the mineral oils the tendency is unimportant, even if it exists at all, for the weak organic acids produced through oxidation of the oil will be much less than those produced during combustion of the hydrocarbons in the fuel. The esters which form the fatty oils, however, can react directly with water or steam at quite moderate temperatures to form alcohols and free fatty acids. Thus, castor oil would react, if used to lubricate the cylinder of a steam engine, according to the equation

$$\begin{aligned} C_3H_5[C_{17}H_{32}OH.CO.O]_3 + 3H.OH \\ = \underset{\text{glycerine}}{C_3H_5(OH)_3} + \underset{\text{ricinoleic acid}}{3C_{17}H_{32}OH.CO.OH}, \end{aligned}$$

and the result would be that the fatty acid set free would corrode the metal of the cylinders. In internal combustion engines some steam is formed by the combustion and this, reacting with the oil on the cylinder walls, may lead to a slow increase of acidity in the oil after prolonged use.

As stated earlier, it is impossible to distil any of the fatty oils without decomposing them. The mineral oils, just like the volatile petrols in the hydrocarbon family, are composed of mixtures of substances which would, if separated, have very different boiling-

points. If heated under atmospheric pressure a lubricating oil would begin to boil at about 400° C., with some decomposition and oxidation, and the boiling-point would rise as the more volatile constituents were distilled off.

In order to reduce the amount of decomposition which takes place when oils are refined by distillation, it is customary to carry out the distillation under a fairly high vacuum; for in this way the initial boiling-point may be lowered to the neighbourhood of 100° C., and the whole range of the oils which constitute the lubricating fractions can be distilled over without heating the crude oil to a point at which the amount of cracking of the large molecules becomes serious. It is essential, in distilling lubricating oils, to reduce the amount of cracking to an absolute minimum, and to this end the use of steam in the refining process was introduced. The steam, superheated to the required temperature, is blown through the oil during distillation and carries away with it the hydrocarbon vapours as they are formed. The vapours are thus swept rapidly away to the condenser instead of remaining at the temperature of the still and having time to undergo decomposition.

ART. 29. *Boundary lubrication.*

Although true boundary lubrication plays only a small part in the normal lubrication of an internal combustion engine there are two reasons why the present chapter would be incomplete without some description of it, and of the experiments on which the theory of it is based. In the first place the two primary films, one on each metal surface, with which it deals, are always there, and although their reactions with one another may only occasionally become critical, because of the presence between them of a comparatively thick film of oil, nevertheless it is upon their behaviour to one another that the danger of a seizure depends, in the event of the full fluid lubrication having failed. The other reason, although less practical, is perhaps no less important. In all the vast number of experiments of all kinds which have been made upon the lubrication of machinery, an outstanding fact is the difficulty of achieving a close repeatability. It arises from the complexity of the controlling conditions and of the chemical nature of the average oils of commerce. The result is that almost all the experiments upon what may be called an engineering scale have led to generalizations which show here and there exceptions and inconsistencies.

Unlike this state of things, the theory of boundary lubrication has been built up on the use of pure chemical substances; the conditions of the experiments can be minutely defined; and the results are

numerically exact, and repeatable to within the very close limits of accuracy of the observations. The theory, as a result, is no more assailable than the other conceptions of molecular physics, and it forms a really solid foundation upon which our ideas of the behaviour of lubricants in practice should be built up. Moreover, there are some results of great practical importance, to be dealt with in the next article, which can only be explained by a knowledge of boundary conditions.

As a preliminary to describing the exact experiments of Hardy and his co-workers, it will be advisable to outline some of the ideas upon which their interpretation is based.

All the effective fluid lubricants, in the first place, are composed of bodies of a large molecular weight, and their molecules tend to be of the 'long chain' type in which there are one or more series of hydrocarbon groups related, for example, like those in one of the saturated paraffin series

$$CH_3-CH_2-CH_2 \ldots CH_2-CH_2-CH_3.$$

Among these saturated paraffins the molecules become larger by the insertion of more and more CH_2 groups in the single chain, the ends of the chain always being terminated by a CH_3 group. In the castor oil molecule there are three long chains, united at one end in the glycerol radicle, thus,

$$\begin{array}{l} CH_2-(CH-CH_2 \ \ldots \ CH_2-CH.OH-CO.O) \\ | \\ CH-(CH-CH_2- \ldots \ CH_2-CH.OH-CO.O) \\ | \\ CH_2-(CH-CH_2- \ldots \ CH_2-CH.OH-CO.O) \end{array}$$

Within each bracket there should be fourteen CH_2 groups in a row, to represent the complete molecule.

It must be noticed in passing that there is a difference between the two types of long chain compounds represented by one of the paraffins on the one hand, and by castor oil or one of the fatty acids on the other, which is of great importance in the formation of surface films. The long chain paraffins, of the general formula

$$C_nH_{2n+2},$$

are perfectly symmetrical, end for end; and moreover, these saturated hydrocarbons are much more stable against chemical reaction than the unsaturated bodies containing a lower proportion of hydrogen to carbon, and than bodies of the series of esters, like castor oil. The reactivity of castor oil and similar unsymmetrical bodies, moreover, is concentrated at one end of the molecule, so that when a film

spreads on a solid surface it is this highly reactive end which is tied most strongly to the surface, leaving the long molecule sticking out like the flexible bristles of a brush.

It is the presence of the (OH) group, or in the case of the fatty acids of the whole (CO.OH) group, which is the symbol of chemical activity, and it is the end of the molecule where these groups are located that is drawn in to a solid surface. The outer end, composed of the more stable CH_3 and CH_2 groups, has only a feeble chemical activity.

Not only are the plain hydrocarbons of the paraffin series wholly

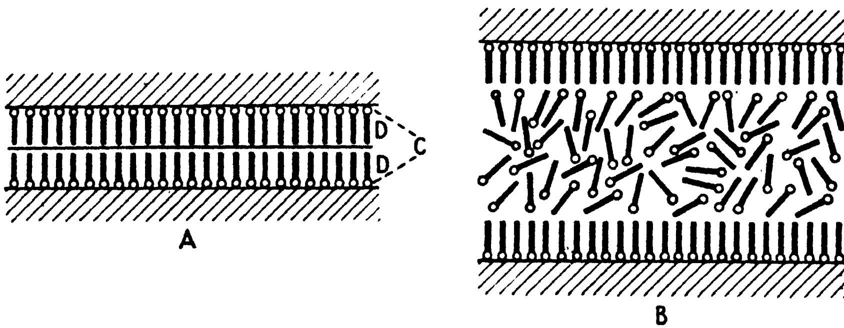

FIG. 45. Polar molecules oriented on a solid surface.

without any of these reactive groups, but, being symmetrical, there is nothing to make them orient themselves in any particular manner in relation to the metal surface. The unsymmetrical molecules with a tendency to orientation are distinguished as 'polar' molecules, and the strength of their polar quality varies with the type and arrangement of their chemically active groups.

The condition of two neighbouring solid surfaces each carrying a mono-molecular layer of polar molecules may be imagined like fig. 45 A, in which the small circles, as at *C*, represent the strongly adsorbed ends of the polar molecules and the straight parts, as at *D*, represent the inactive hydrocarbon chains. Fig. 45 B represents the same surfaces separated by a thick film of the same lubricant, and will be referred to again below.

It must not be thought that the use of the structural formulae given above, or the references to the 'ends' of a molecule, are unjustifiable flights of fancy. Although devised originally to summarize and explain chemical behaviour, the molecules they represent have since been shown by physical measurements to conform in space to such schematic arrangements. The lengths and something which corresponds to the cross-section of the different molecules can now

be stated with confidence. From measurements of the spreading of their films on a clean water surface Langmuir[17] has calculated the lengths and cross-sections of a number of large molecules, and some typical figures are given in table 22.

TABLE 22

The dimensions of some fatty acid molecules and related compounds.

Substance	*Cross-section sq. cm.*	*Length cm.*	$\sqrt{(\textit{Cross-section})}$ *cm.*
Palmitic acid . . .	21×10^{-16}	24×10^{-8}	$4{\cdot}6 \times 10^{-8}$
Stearic acid . . .	22×10^{-16}	25×10^{-8}	$4{\cdot}7 \times 10^{-8}$
Oleic acid	46×10^{-16}	$11{\cdot}2 \times 10^{-8}$	$6{\cdot}8 \times 10^{-8}$
Tri-stearin	66×10^{-16}	25×10^{-8}	$8{\cdot}1 \times 10^{-8}$
Tri-olein	126×10^{-16}	13×10^{-8}	$11{\cdot}2 \times 10^{-8}$

It will be seen that the molecules are all of an elongated shape. If one takes the square root of the cross-sectional area as representing a sort of average diameter, then the length of the palmitic acid molecule is 5·2 times its diameter and that of oleic acid about twice. Peculiar interest attaches to a comparison between the dimensions for tri-stearin and stearic acid and between tri-olein and oleic acid, for tri-stearin and tri-olein each have molecules very like castor oil, with three parallel chains, each of the chains being identical with the corresponding acid molecule, less one hydrogen atom. The measured length of each acid molecule is almost the same as that of its derivative, but the cross-sections of the tri-olein and tri-stearin molecules are each just three times those of the corresponding acid, as they would be if the three chains in the molecule lie close and parallel to one another.

In any attempt to measure the friction between two surfaces in a true state of boundary lubrication the first essential is to obtain the surfaces in an absolutely clean condition, and the second is to be sure that the surfaces are separated only by their primary films and not by a free film of lubricant of finite thickness. The starting friction between two clean dry surfaces under a given normal pressure can be reduced to less than half by a layer of lubricant one molecule in thickness, so that quite invisible traces of any foreign matter will completely upset the observations, and it has been the failure to start with chemically clean surfaces which has led to so many discrepancies between lubrication experiments in the past.

When once the surfaces *are* clean, extremely interesting and consistent results are obtained. A spherically shaped slider, loaded and resting on a plane surface, will at once sink through a pool

of oil until the spherical and flat surfaces are separated at their very minute area of contact only by the primary films attached to each. Hardy,[18] using such a slider and measuring the friction by the horizontal pull necessary to start the slider from rest, experimented with a number of pure chemical substances as lubricants, in conjunction with sliders and plates of different materials, glass, steel, bismuth, etc. He first showed that the coefficient of friction is truly

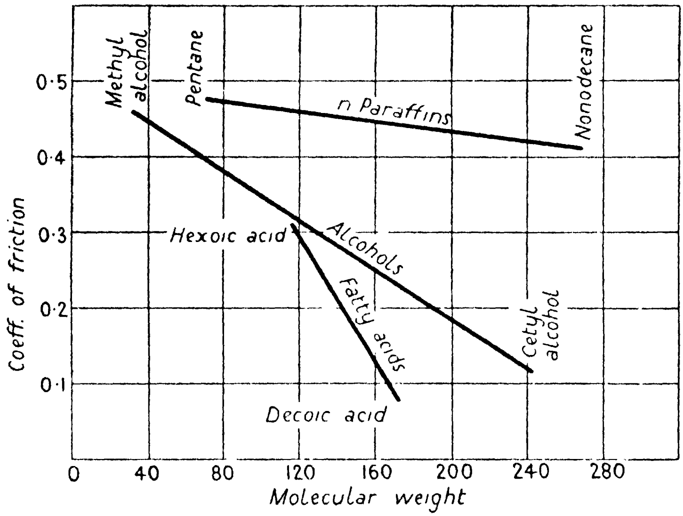

FIG. 46. Relation between coefficients of friction (for steel-on-steel) and molecular weights of certain pure chemical substances used as lubricants.

independent of the load when this is above a small limiting value, and then observed the coefficients for three series of organic compounds. Although many of these would not ordinarily be regarded as lubricants at all, the experiments showed that they did in fact behave as such, and that their properties were exactly related to their molecular weights.

Only the results for a steel plate and slider are given in fig. 46, but these are typical in character for all the materials used. The three straight lines show how the coefficient of friction was found to vary for each series of chemical compounds in an exactly linear manner with the molecular weight. The three series of liquids used were

(*a*) paraffins, of general formula C_nH_{2n+2} between C_5H_{12} and $C_{19}H_{40}$;

(*b*) alcohols, of general formula $C_nH_{2n+2}O$ between $CH_3(OH)$ and $C_{16}H_{33}(OH)$;

(*c*) acids, of general formula $C_nH_{2n}O_2$ between $C_5H_{11}(CO.OH)$ and $C_9H_{19}(CO.OH)$.

It will be seen that the symmetrical and non-polar paraffins show the highest coefficients of friction, and the ones varying least with molecular weight. The molecules both of the alcohols and the acids contain the active (OH) group, while, of the two, the acids are the more chemically active and hence the more ready to orient themselves in a regular manner on the solid surface. The coefficients of friction are correspondingly lower for the more strongly polar molecules of the acids, and they show, moreover, so rapid a fall with molecular weight, that at a value of about 190 the coefficient should be zero. The straight lines indicate that in each series a molecular weight can be reached above which the friction is zero, and experiment showed that in those circumstances the smallest tractive force that could be applied did in fact always produce a slow slip.

A change of the material of the slider and plate had not the smallest effect upon the slope of the line for any one type of compound, but shifted it bodily up or down by a definite amount.

The coefficient of friction can be completely expressed in the form

$$\mu = b - d - c(N-2),$$

in which b depends only upon the nature of the solids of which the slider and plate are made; d depends only on the chemical series to which the lubricant belongs; N is the number of carbon atoms in the chain, and c is the decrement of the friction due to each carbon atom, which depends upon the chemical series. It will be noticed that the expression for μ contains no temperature term, for as soon as boundary lubrication with a pure substance is established the force of friction is found to be entirely unaffected by temperature over the range explored, namely from about 15° to 106° C.

Other striking facts observed when a state of boundary lubrication has been established are, that with a given lubricant the friction is the same whether the temperature is above or below its melting-point; that for any given chemical series the straight lines connecting μ with the molecular weight, as in fig. 46, show no change in direction when increase of the molecular weight alters the lubricant, at a given temperature, from a fluid to a solid; and that when the friction between two initially clean surfaces is measured in a closed vessel containing only the *vapour* of a lubricant, the reduction of friction below the clean surface value is proportional to the vapour density in the containing vessel. It may fairly be argued that the number of adsorbed molecules on the solid surfaces will be proportional to the vapour density, and hence that when the vapour of a lubricant is allowed gradually to condense upon the clean surfaces of the slider and plate each molecule contributes independently to the reduction

of the resistance to motion. Finally, it has been shown that the value of the friction obtained between the slider and plate when the primary film has been allowed to form from an atmosphere of the *saturated* vapour is identical with that observed when the slider is standing in a pool of the liquid lubricant. We have here, therefore, very strong evidence that the full reduction of friction below the clean surface value can be obtained by a primary film only one molecule thick.

ART. 30. *Boundary layer friction with compound lubricants.*

So long as the force of friction depends upon the viscosity within a fluid film of finite thickness it must, of course, vary considerably with the temperature, and the existence of a state in which friction is independent of temperature is therefore a crucial test that a condition of boundary lubrication exists.

There may, however, be a condition of boundary friction without independence of temperature if the lubricant is not a single chemical substance, but a mixture of different molecules, and some most important practical deductions can be made from observations upon simple mixtures of lubricants.

The horizontal line near the top of fig. 47 shows the observed values of the coefficient of friction, μ, under boundary conditions between 20° and 106° C. for the steel slider and plate when lubricated by the familiar colourless medicinal paraffin oil. This oil, for medical purposes, is rendered as completely inert as possible from the chemical point of view, by the extraction of all the unsaturated and otherwise active elements. From what has been said in the last article it will be understood that a complete removal of unsaturated and polar molecules would reduce the lubricating value of an oil under boundary conditions to a minimum, through the loss of all those components which are strongly adsorbed, and hence that the addition to such an oil of a second, consisting of strongly polar molecules, might be expected to produce a fall in the observed coefficient of friction.

Palmitic acid is a fatty acid with a strongly polar molecule of molecular weight 256 and composition

$$C_{15}H_{31}CO.OH.$$

It is one of the substances above the critical molecular weight at which the straight line for its series (see fig. 46) would cut the axis of zero friction, and when experimenting with it Hardy did, in fact, find that the smallest force he could apply to the slider produced a slow movement.

The lower curves in fig. 47 exhibit the observed effect upon μ of small additions of palmitic acid—no more than 0·014 per cent., 0·33 per cent., and 0·97 per cent.—to the normal B.P.* paraffin. It will be seen that above temperatures ranging from 30° to 50° C. μ becomes independent of the temperature at a value very substantially below the value 0·23 observed with the B.P. oil alone.

Now palmitic acid is a solid with a melting-point at 63° C., and at room temperature it is not readily soluble in B.P. oil. The curves

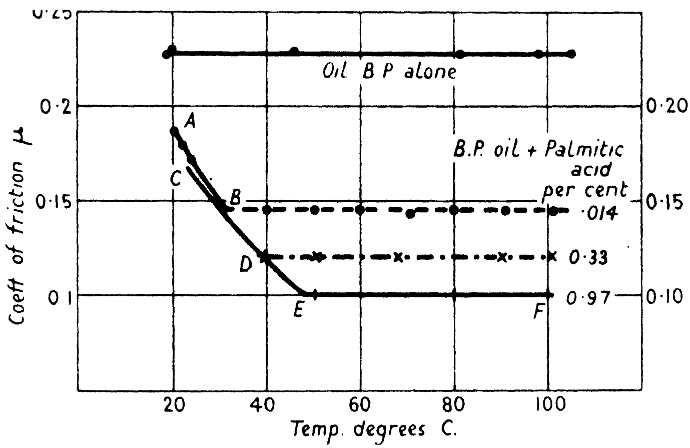

FIG. 47. Friction-temperature curves for paraffin oil B.P. and for the same oil with additions of palmitic acid.

show that a condition of equilibrium is reached at a certain temperature, above which the friction is constant, and that up to additions of 0·97 per cent. the maximum lowering of the friction depends upon the amount of palmitic acid added. Higher amounts were tried, up to 6 per cent., but the curves obtained were identical with that for 0·97 per cent.

The temperature ranges of falling friction, *AB* and *CDE* on the curves, clearly represent a series of conditions in which the palmitic acid is becoming more and more dominant in the boundary layer, owing to its increasing solubility in the oil with rise of temperature and to its readiness to form an adsorbed film which displaces the non-polar B.P. oil. At an addition of 0·97 per cent. complete solubility is only reached near the melting-point of the acid. At lower percentages complete solubility, and a constant composition of the adsorbed layer, is reached at lower temperatures, but the concentration of palmitic acid molecules is not sufficient to produce the full

* Prepared according to the British Pharmacopoeia.

effect in lowering the observed friction. We may suppose that at 0·97 per cent. the acid is sufficient to monopolize the primary film completely, to the exclusion of all non-polar molecules, and thereafter further additions of acid give curves identical with *CEF*.

A close counterpart of these results in the sphere of practical engineering is to be found in the improvement of the lubricating qualities of mineral oils obtained through the addition of small quantities of a fatty acid. This effect was clearly demonstrated by

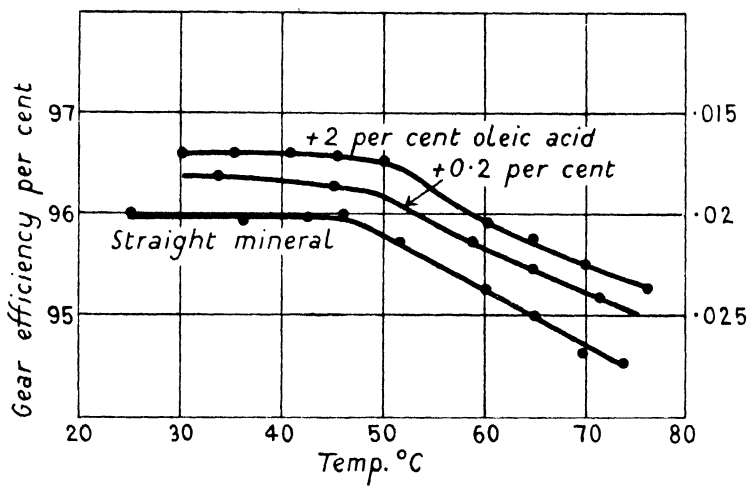

FIG. 48. Efficiency-temperature curves for Lanchester worm-gear machine with a straight mineral oil and with the same oil plus additions of oleic acid.

Hyde[19] in experiments on an engineering scale at the National Physical Laboratory using a worm-gear testing machine. This machine enabled an accurate measure to be made of the efficiency of a set of worm-gears while transmitting power at different known temperatures. The gear-teeth operated under very heavy loads and with mixed conditions of boundary and viscous friction. The proportion of the power being wasted in boundary friction was sufficient to show a 10 per cent. reduction of the total loss when 0·2 per cent. of oleic acid was added to a straight mineral oil.

The curves of efficiency for the gear, and of a calculated mean coefficient of friction, between 25° and 75° C. are shown in fig. 48. The addition of 0·2 per cent. of acid produced an increase of about 0·4 per cent. in the efficiency throughout the temperature range, while adding 10 times that amount produced only a further 0·2 per cent. in the efficiency.

The fall of efficiency and increase of friction with temperature is

no doubt to be explained by the fact that as the oil became less viscous it was more rapidly squeezed out from between the gear-teeth, so that larger and larger proportions of the normal forces between the gear surfaces were taken by direct contact between the primary films. The increase of friction from this cause would more than balance some reduction of the loss due to viscous friction, for it must be remembered that the average coefficients of friction obtained with fluid film lubrication are much lower than those for boundary friction, even with the best lubricants.

It is not possible to give definite figures for the coefficients of kinetic friction between two primary films. Hardy's observations were all upon the force necessary to start his slider from rest, and they yield, therefore, coefficients of static friction. The lower limit of μ for a mineral oil in these static experiments may be taken as being roughly 0·1, but there is evidence of a fall in the friction as soon as movement occurs. The most likely thing appears to be that under kinetic conditions the slipping over one another of the real primary monomolecular layers ceases to hold, and that the friction rapidly develops into that produced by the slipping of layers several molecules thick, the outer layers being oriented in a more or less orderly manner when the molecules of the lubricant are strongly polar, but not so firmly held by the chemical bonds of the solid surface as the strongly adsorbed primary layers. Under kinetic conditions, therefore, the physical state of the surfaces is altered, and can be less completely defined than in Hardy's static experiments. Boundary friction under kinetic conditions will shade off into fluid film lubrication without any clear line of demarcation between the two.

Fig. 45 B (p. 110) was meant to give some idea of the state of things between two neighbouring surfaces, each with a strongly adsorbed primary layer, and the upper surface showing also a weakly adsorbed and oriented secondary layer. Between this secondary layer of the upper surface and the primary layer of the lower is a film about 3 molecules thick in which there is no orientation and the molecules are arranged indiscriminately. Within these layers the heat motions of the molecules are unaffected by the fields of force from the solid surfaces.

In the worm-gear experiments represented in fig. 48 the load is transmitted at either line or point contacts, and one would expect films of all thicknesses down to a very few molecules at the points where the load is concentrated. This is borne out by the value of the estimated average coefficient of friction for the gear, 0·020 to 0·027 for the straight mineral oil (see fig. 48). Under conditions of static boundary friction it might be expected to be about 0·1, and with complete fluid lubrication to be of the order of 0·003 at a speed of

1,000 ft. per min. In fluid friction there is, of course, no coefficient in the ordinary sense, the friction force being dependent on the speed of motion and the area of the rubbing surfaces.

There can be little doubt that the lowering of the apparent coefficient of friction by about 0·002 in the worm-gear testing machine when 0·2 per cent. of oleic acid was added to the mineral oil is to be explained by surface changes of the same kind as those when the static coefficient in fig. 47 was lowered from 0·23 to 0·10 by the addition of 0·97 per cent. of palmitic acid. In each case the 'oiliness', or lubricating property under boundary conditions, of a non-polar oil was substantially improved by the addition of a very small proportion of polar molecules. These form strongly adsorbed primary layers on the solid surfaces, displacing the non-polar molecules and reducing the coefficient of friction accordingly.

These small additions of less than 1 per cent. to an oil will have no appreciable affect upon its viscosity and will not, therefore, influence the friction losses of an engine during normal running. Although oiliness is an advantage in a lubricant for internal combustion engines, chemical stability is even more important; and there is some evidence that the addition of fatty acids and fatty oils, even in small percentages, for improving the oiliness of a mineral oil has an adverse effect upon its tendency to form carbonaceous products in the cylinder. For this reason such additions need to be made with caution in high duty engines with high cylinder and piston temperatures.

The tendency of an oil to form carbon and sludge, and the relation of this tendency to its chemical composition, will now be considered in the next two articles.

ART. 31. *Carbon and sludge formation.*

If a clean engine is filled with clean oil and run for even a comparatively few hours at a high power-output, not only will the oil have become black, but the crankshaft and connecting-rods will, with a mineral oil, be found covered with a black surface film which when cold is quite hard and takes on a polish like varnish. After prolonged running a sticky black sludge will be found separated out from the oil at certain points, such as inside a hollow crankpin, and there will be caked carbon on the top of the piston. It may be added for later reference, moreover, that as a general rule the blackening of the oil and the production of a filthy condition in the engine is more rapid in engines of the Diesel type than in a properly adjusted petrol engine.

Clearly the blackening of the oil must be due to the presence of suspended carbon in a finely divided state: so fine that much of it

cannot be removed by filtration. What causes the carbon formation, and where it is produced, will be discussed later. A further essential element in the production both of sludge and of hard carbon is the formation of gummy substances by the polymerization and partial oxidation of the oil molecules, the latter process being one which is much accelerated by the presence of a metal surface. Probably some gum formation begins at once, but the early products of oxidation are soluble in the oil itself, and an engine may run for very many hours after the oil has become perfectly black with suspended carbon without the amount of gum formed being serious or even perceptible, unless the temperatures throughout the engine are exceptionally high. If they are, decomposition and oxidation will be more rapid and the more viscous products of the process will collect, already black with carbon, in such places as piston-ring grooves, drain-holes, and in the wider parts of the oil-ways in the crankshaft where the flow is sluggish. At first the decomposition products would only show increased viscosity, but under the influence of sluggish movement and a sustained high temperature their nature gradually changes to that of something sticky or nearly solid, and quite black.

At this stage they form the *sludge* which is found collecting in oil-ways and in the drain-holes from the ring grooves in the piston. Wherever collections of sludge are constantly in contact with more oil, as inside the crankshaft and over the lower parts of the piston, it may go on gathering to itself more sludge, but it has no opportunity of getting much beyond the sticky or india-rubbery condition. Any sludge, however, which may have collected in the upper piston-ring grooves or round the top land of the piston, is subject to far more severe temperature conditions, and in an overheated engine may rapidly congeal to a hard cement in which the rings become fixed, and piston seizure is then imminent. The hard cement found round the top land and on the piston top consists very largely of carbon and, together with the similar products on the valves and cylinder heads, represents the amount of *carbonization* which has taken place.

Some carbonization there will always be, on the piston top and the walls of the combustion space, even when there is no danger at all from stuck rings; but here again the final product is the result of decomposition and partial oxidation: a high temperature acts first on the oil to form gums impregnated with carbon, and then turns these into a black solid, hard or soft, dry or oily, according to the design of the engine and the adjustment of the oil supply. Most carbon deposits have probably gone through an earlier stage when their nature was that of a not insoluble gum impregnated with soot from

the combustion, although the progress to the final, hard, insoluble deposit, being continuous and yet under different conditions from point to point, it would be impossible to separate it into definite stages.

It may be objected that dry and flaky carbon may sometimes be found on the piston-top and cylinder-head surfaces with no sign of gummy or caked products. This is, indeed, the ideal to be aimed at. We can scarcely hope to avoid soot, and the consequent blackening of the oil, for reasons explained below, but so long as this is not associated with gum formation to consolidate it into fixed carbon there will be no danger to the piston, nor will the carbonization on the piston top be serious.

It has been suggested by Thornycroft and Barton[14] that the amount of carbonization with any given oil depends entirely upon the behaviour of that oil in the combustion space above the piston. Partial combustion of small droplets of oil thrown off the piston at the top of the compression stroke is put forward by these writers as the cause both of carbon and gum formation, and figures are given of the measured weight of carbon on a piston top after 50 hours running which go to show that only when the petrol-air mixture was rich did the combustion of the fuel appear to contribute to the weight of carbon collected.

Undoubtedly most of the finely divided carbon is formed on the combustion side of the piston, and possibly much of the gum also, but an alternative picture of what happens, given below, according to which the soot and the gums that together make for carbonization are of separate origin, appears to fit all the facts rather more readily. Although it has been shown that oil is thrown off from a piston in a fine spray when an engine is motored at a high speed with the cylinder heads off, it must be remembered that engines can be run for long periods with so low an oil consumption as to make it almost inconceivable that oil could be sprayed from the pistons. A 12-cylinder aero-engine, for example, of bore and stroke $5 \times 5{\cdot}5$ in., has been run for 100 hours at 2,250 r.p.m. and $\frac{9}{10}$ full load with an average oil consumption of $2\frac{1}{2}$ pints per hour. A simple calculation shows that this is equivalent to just about 1 cu. mm. of oil per cylinder per rev. This oil spread over the cylinder bore would make a film 2×10^{-6} cm. thick, so that it only needs a film of this ultra-microscopic thickness to be burnt off or evaporated per stroke to leave nothing to be thrown off by the piston.

There are good grounds for supposing that a certain amount of finely divided carbon, sufficient to produce the blackening of the oil, is derived from imperfect combustion of the fuel. Within a thin boundary layer near the comparatively cool walls (150°–200° C. at the

most) it seems certain that complete combustion of the fuel-air mixture must be arrested, and some carbon thrown down. This is sufficient to explain the presence of some finely divided carbon in the cylinder-gases during every working stroke. Swept in all directions by gas movements, some of this will stick to the cylinder walls on which many of the surface molecules of the oil film, by the end of the stroke, must have been decomposed. During the exhaust stroke the sooty and half-decomposed surface of the oil is first overrun by the piston and then washed by fresh oil thrown up by the connecting-rod big end.

This appears to be an adequate explanation of the rapid blackening of the oil in the crankcase, and in being quite independent of any gum formation it fits in with the fact that blackening of the oil can take place in a cold engine at moderate speeds, when gum formation would be negligible and the oil too viscous to the thrown off the pistons. The more rapid fouling of the oil in a Diesel engine follows naturally, for in that type the soot in the cylinder gases is sufficiently prevalent to be visible in the exhaust, at all except light loads.

There is, however, no reason to suppose that *all* the carbon produced is formed first above the pistons. The black varnish found with mineral oils on the connecting-rods and on the under sides of the pistons may very well owe its origin to oxidation of the oil at the metal surfaces where it is found. Steel and aluminium are known to be active catalysts for the oxidation of mineral oils, and the fact that a similar blackening is not found with castor oil, although the oil itself becomes no less black, suggests that the deposits obtained with mineral oil are due to local break-down of the oil. The cleanliness of the metal surfaces with castor oil is also to be associated with the acidity which accompanies the decomposition of the fatty oils, as explained in the last article. The fatty acids formed, if any moisture is present, have a faintly corrosive and hence a cleansing effect on the metal surfaces.

Turning now to the question of gum formation, a proportion of the scorched and decomposed surface film will find its way into the ring grooves of the piston during the exhaust stroke, and if allowed to remain there at a high temperature it will provide the material for a gradually stiffening black cement. The early products being soluble in the oil itself, a very ample oil supply should prevent the amount of gum formed ever becoming serious, even in a very hot engine. This has in fact been found to be the case, and it is a result which it would be difficult to reconcile with the suggested spraying of oil from the piston as what leads to gum formation; for this could then scarcely fail to become worse and worse the more oil there was

to be thrown off, whereas if the gum is first formed on the cylinder walls an ample oil supply would only facilitate solution of the gum without increasing its amount.

In the crankcase the oil will be sprayed in all directions in an oxidizing atmosphere, but the temperature will rarely be above 150° C., and with the oils in use for aero-engines the rate of oxidation at this temperature is very slight, even under the most favourable conditions. A certain amount of the semi-oxidized products will be washed from the cylinder walls into the crankcase oil and will be ready to form sludge at any points where it can collect, but apart from the formation of the black varnish on the metal surfaces, already referred to, there seems no reason to suppose that oxidation proceeds in the crankcase to any considerable extent except on the under sides of the pistons.

For a further discussion, and some recent experimental results, on the subject of sludge formation, a paper by Barnard and others[71] may be consulted.

ART. 32. *Carbonization in relation to chemical composition.*

The blackening of a lubricating oil by soot formed from the fuel being, in itself, harmless, the essential factor in carbonization is the liability of an oil to form partial oxidation products of a gummy or resinous nature; and the best oil from this point of view will be the one which is most stable against oxidation or polymerization at the working temperatures.

The great difficulty in the way of correlating an oil's resistance to oxidation with its chemical composition lies in the lack of real knowledge of what the composition is. Between the broad classes of the fatty oils and the mineral oils there are wide differences of behaviour and composition, and the chemistry of the fatty oils is known with some exactness. As between different mineral oils, however, one has to be content with describing them as Pennsylvanian 'bright stock', a 'naphthenic base' oil from Russia, or an 'asphaltic base' from Venezuela, the names signifying no more than the source and the type of crude oil from which they come. The whole problem would be vastly simplified if the components of the mineral oils could be segregated from one another and if the stability of the various constituents in the presence of oxygen could be experimentally assessed. In the absence of any such analysis the mixed oils of commerce must be treated as a whole, and their reaction to oxygen must be assessed indirectly by physical tests of viscosity and kindred properties. Such physical tests have been widely used for specification purposes and will be referred to again

later on. They provide a fair general indication of the chemical stability of an oil, but the conditions under which oxidation takes place are very different from those in an engine, and it cannot be claimed that the behaviour of an oil in an engine as regards carbonization can be foreseen with certainty by any laboratory oxidation test so far devised.

The rate of absorption of oxygen by lubricating oils at different temperatures has been directly observed by Mardles, using a method in which 1 gm. of the oil was enclosed in a small glass cylinder 3 cm. in diameter and 150 c.cm. capacity, filled with air. The glass cylinder was rotated slowly inside an electric furnace at a known temperature, and the oxygen content of the air was measured after various intervals of time. The results were then expressed as so many milligrams of oxygen absorbed by a gramme of oil in 1 hour.

Without knowledge of what happens to the absorbed oxygen, and whether the chemical changes brought about by it are not very different in different oils, the test can never be a conclusive one of an oil's behaviour in an engine. And there is evidence, moreover, that an oil which in this form of test does not readily absorb oxygen may yet show up badly in regard to carbon deposits during an engine test. The oxygen absorption test is nevertheless of some interest in the striking way it differentiates between castor oil and the mineral oils; and it is of value also as a way of showing up the catalytic action of certain metals in promoting oxygen absorption, and its prevention by inhibitors.

In fig. 49 are given the results of the tests upon castor oil and three different types of mineral oil. It is at once evident that castor oil is in a different class from the others. It readily absorbs oxygen at temperatures above 125° C. and rapidly increases in viscosity. Under the conditions of the experiment 1 gm. of castor oil at 150° C. absorbed about 20 mg. of oxygen in 1 hour. None of the mineral oils absorbed oxygen appreciably below about 140° C., and even at 250° C. the Pennsylvanian bright stock after heating for an hour had only absorbed about 10 mg. of oxygen. After 3 hours heating at 250° C. the gummy and oxidized oil was still soluble in benzene without residue.

The Russian oil behaved in a very similar way, but appeared to be a little more ready to absorb oxygen at 150° C. and above, although it showed signs of better stability below that temperature. This is shown by the dotted curves on the right of fig. 49 which show the oxygen absorption, to a more open scale, at temperatures below 140° C. There was no appreciable absorption by any of the oils except castor oil below 110° C.

The asphaltic base oil was definitely inferior to the other two mineral oils, the rate of oxygen absorption at 175° C. being about the same as that of the Pennsylvanian oil at 250° C. This difference between asphaltic oils and those of naphthenic or paraffin base is generally in accordance with experience on engines.

The presence of strips of copper, iron, and aluminium placed in

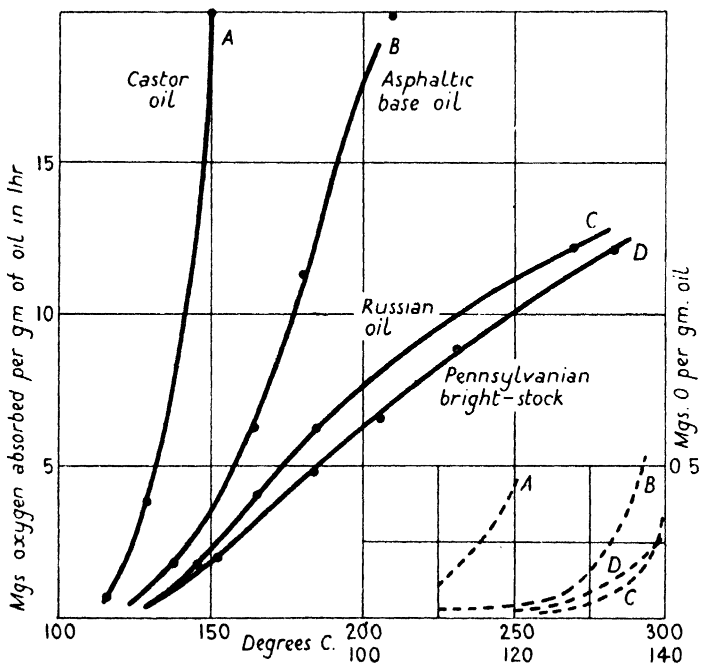

FIG. 49. Rate of absorption of oxygen by castor oil and three different mineral oils when maintained at various constant temperatures.

the oil did not increase the rate of oxygen absorption by the mineral oils below 175° C., but tended to make the oxidized oil discoloured, thick, and insoluble in benzene. From the point of view of piston-ring gumming, therefore, a cast-iron piston ring in an aluminium piston may have a very detrimental effect upon the partially oxidized oil which is scraped from the cylinder walls into the ring grooves. At temperatures above 175° C. the metals, especially iron, increased the rate of oxygen absorption. When the metals were added in the the form of filings or thin foil their catalytic effect was much more marked. Iron filings, for example, were found to increase the rate of oxygen absorption of several oils between 125° and 250° C. by as much as eightfold, and aluminium foil as much as threefold, when 0·1 gm. of the metal was added to 1 gm. of the oil.

Many attempts have been made to inhibit the oxidation of oils in the presence of air by the addition of small quantities, of the order of 1 per cent., of 'stabilizers' such as cresol, β-naphthol, &c., which have shown beneficial results in preventing gum formation in volatile spirits containing unsaturated hydrocarbons. Their beneficial action in volatile spirits would naturally be at temperatures not above 20°–30° C. When tested with oils by the oxygen absorption method some of them were found to cause a reduction of the initial oxidation rate, but after an hour or so the effect became less and ultimately disappeared, probably as a result of the oxidation of the stabilizers themselves.

To set against the favourable showing of Pennsylvanian bright stock in fig. 49, Thornycroft and Barton have given the results of a series of 50-hour duration tests on a water-cooled engine which indicate that the addition of bright stock to a blend of mineral distillates (see art. 28) increases the amount of carbon on the piston. Some of these results are given in table 23. The oils were prepared

TABLE 23

Carbon formation with mineral oil distillates and blends with bright stock.

	Blend of distillates	*Distillates 80 per cent. bright stock 20 per cent.*	*Distillates 60 per cent. bright stock 40 per cent.*
Oil consumption in 50 hours, pints	1·50	1·45	1·75
Carbon collected from the piston crown, gm.	0·43	0·90	1·43
Viscosity of unused oil by Redwood I. at 140° F. . .	175	175	190
200° F. . .	58	60	66

from refined distillates of the same origin, and the proportion of low and high viscosity distillates were adjusted so that the different blends with bright stock had approximately the same viscosity at 140° F. (60° C.). The results must be accepted with caution, not only because of the inherent difficulty of getting repeatability in such tests, but because, as the authors themselves point out, bright stock oils from different sources vary considerably in their carbon-forming propensities.

The greater danger with the fatty oils of getting hard and gummy carbon deposits on the pistons (as distinct from a dirty appearance of the crankcase parts) is now generally accepted. It is only what might be expected from oils which are first cousins of the 'drying oils', and themselves absorb oxygen so readily at temperatures well

below those prevailing in the piston. The figures in table 24, also taken from Thornycroft and Barton's paper, show that the addition of 5 per cent. of a fatty oil to a mineral oil distillate caused an increase of rather more than 50 per cent. in the amount of carbon deposit during a 50-hour run.

TABLE 24

The effect on carbon formation of fatty oil added to a mineral distillate.

	Mineral oil distillate	*Mineral distillate + 5 per cent. fatty oil*
Oil consumption in 50 hours, pints . . .	1·55	1·45
Carbon collected from the piston crown, gm.	0·41	0·63

That a quite small addition of a really badly gumming oil should have a large effect upon the carbon deposits is only to be expected, when one considers how small a proportion of the oil consumed is actually concerned in forming the carbon deposits. During the 50-hour runs quoted in tables 23 and 24, for example, the amount of oil consumed was from 1·45 to 1·75 pints, or 650–800 gm., while the total carbon deposits varied from 0·41 to 1·43 gm. They varied, therefore, from 0·06 to 0·18 per cent. of the weight of the oil consumed. This constitutes a serious difficulty which affects the reliability of single-cylinder engine tests for classifying oils, for clearly quite a small change in the conditions of the test might bring about a large alteration in the weight of the deposits collected with any given oil, when so large a proportion passes out through the exhaust valve in a half-burnt condition.

There are chemical factors, too, unconnected with the type of oil as defined by its origin, which have been shown to affect the amount of sludge and carbon produced, to a degree quite as great as changes in the oils themselves. Thornycroft and Barton have given the results of two 50-hour tests using oils of the same viscosity, from the same asphaltic crude, in which a difference of the refining process led to carbon deposits at the end of one test nearly double those in the other.

Other tests at Ricardo's laboratory have shown that during a 50-hour run in which the oil was circulated continuously, and the sludge formed in the oil was collected in a centrifugal separator during the progress of the test, the amount of sludge was increased as much as fivefold when ½ per cent. of water was added to the oil after every 8 hours of running. Five different oils were tried, and the smallest increase in the sludge caused by the water was 33 per

cent. in the 50 hours. Additions of water larger than ½ per cent. in 8 hours produced almost the same results, which were satisfactorily repeatable.

Although no water would normally be present in the lubricating system of an aero-engine, there is undoubtedly the possibility of the condensation of water from the burnt gases during the starting-up period, and of this water reaching the oil in the crankcase. The quantity no doubt would be small, but might be sufficient to alter appreciably the sludging of the oil.

ART. 33. *Laboratory tests of chemical stability.*

In view of the uncertainties referred to in the last article a prolonged test of 100 hours running or more, in the actual engine for which it is required, is the only conclusive test of how an oil will behave in service. Such prolonged and expensive tests are ruled out for specification purposes, and failing them a laboratory oxidation test appears to be the best compromise. While admittedly tests of this type cannot reproduce engine conditions, they do at least give repeatable results; they can be performed in the apparatus of a chemical laboratory; and they can at any rate be trusted to exclude any really bad oils, even if they may on occasion be unfair to some good ones.

A widely used form of oxidation test, or combination of tests, is one in which the oil is maintained at a fixed temperature while air is bubbled through it at a controlled rate. In the British Air Ministry specification test* for mineral lubricating oils, 40 ml. of the oil are placed in a glass tube 10 in. long by 1 in. internal diameter, and this is maintained at a temperature of 200° C. while air is blown through it at the rate of 15 litres per hour for two periods of 6 hours on consecutive days. The viscosity of the oil before and after the 'blowing' is determined at 100° F. and the oil is rejected if the viscosity at the end is more than 2·0 times that the beginning.

In this test the physical property of viscosity, which is of doubtful importance in actual service, is used as an indicator of the chemical stability of the oil. The test assumes that an increase of viscosity is necessarily correlated with the formation of the gummy oxidation products which it is desired to exclude as far as possible. The assumption is no doubt justified in relation to the oxidation test itself. The weakness of the test lies in its failure even remotely to reproduce the conditions of service in an engine, and in the necessity for close temperature control if repeatable results are to be obtained. A difference of 1° C. in the blowing temperature of 200° C.

* Specification No. D.T.D. 109. H.M. Stationery Office.

may cause a substantial alteration in the increase of viscosity during blowing.

This oxidation test is associated in the same specification with a 'coking' test designed to show up those oils which contain in suspension a high proportion of asphaltic and resinous substances, readily converted by heat into carbonaceous deposits. In the coking test about 4 gm. of the oil is enclosed in a weighed cylindrical glass bulb, drawn out into a short capillary at the top end, and is subjected to a temperature of 550° C. by immersion in a bath containing molten metal, the glass coking bulb being enclosed in a close-fitting cylindrical steel sheath.

The coking bulb is allowed to remain in the bath for 10 min. after the fuming of the oil has ceased, and the amount of asphaltic and resinous matter remaining is then determined by weighing, after the bulb has cooled. The weight of the residue, expressed as a percentage of the weight of the oil taken, is the 'Coke Number'. The specification lays it down that when the experiment is carried out according to the directions given, the coke number of the fresh oil shall not exceed 0·65, and furthermore that after the oil has been 'blown' in the oxidation test, the coke number then shall not exceed the original coke number by more than 1·0.

By such a combination of physical tests a check on the chemical stability of an oil, both against oxidation at 200° C. and decomposition at the higher temperature, is aimed at, and to some extent achieved. Undoubtedly there are examples of oils which would fail to meet the specification and yet might stand up excellently during prolonged tests in service. An oxidation test at one particular temperature of 200° C. cannot hope to cover engine conditions which at some points not only exceed 200° C. but reach a figure at which cracking begins, while yet the bulk of the oil may never rise above 150° C.

Other points of weakness in the oxidation test are that it depends upon decomposition in the presence of air, whereas in an engine the oil is subjected to its most severe temperature conditions, round the piston rings, in an atmosphere deficient in oxygen; that it makes no allowance for the effect of the metal surfaces upon the oil; and that it endeavours to bring about in a short time, by more severe temperature conditions, effects which in practice would normally take very much longer to produce.

In spite of these weaknesses, however, one returns to the essentials of a specification test, that in a laboratory the conditions under which it is to be carried out *can* be laid down exactly and, with care in execution, repeatable results are obtained. Single-cylinder engine tests, while approaching, although still imperfectly, the conditions of

service, introduce factors much less easily controlled; not to mention the added complication and cost of the apparatus required, or the fact that no results whatever can be expected from an engine test of less than 50 hours duration.

A mechanical test which is accurately repeatable, and which at the same time carries the test of an oil to the point of actual break-down, is the 'seizing temperature' test developed by Stanton.[20] In this test a steel shaft is rotated in a bronze bush which is loaded and balanced in such a way that the frictional torque between the shaft and the bush can be accurately measured, and any rapid increase

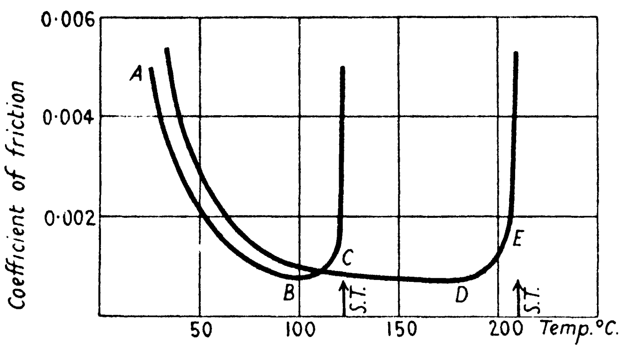

FIG. 50. Friction-temperature curves for a mineral oil in a journal bearing.

of friction caused by a break-down of the shaft-to-bush lubrication can very quickly throw off the load, and prevent damage at the working surfaces.

The working temperature of the oil film between the shaft and the loaded journal can be steadily raised by supplying heat at the centre of the shaft. When this is done under conditions of constant load and speed the friction torque shows at first the steady drop illustrated in fig. 50 from *A* to *B*, due to the diminishing viscosity of the oil. For each oil, however, there comes a point when further heating leads to a rise of the friction torque at a rapidly increasing rate, as illustrated by the portions *BC* and *DE* of the two typical temperature-friction curves. The rising part of the curves does not indicate any unstable condition. If the temperature during the test *ABC*, for example, having reached 115° C., were then lowered to 100° C., the friction would diminish again from 0·001 to 0·0008; but if the heating were continued, the rise of friction would soon become so rapid as to amount to seizure.

The temperature at which the friction-temperature curve for any given oil becomes nearly vertical, called the seizing temperature (S.T.), is repeatable, provided extreme caution is employed in the

preparation of the shaft and the bush surfaces. It can therefore be used as a measure of the quality of the oil. What this quality depends upon, of being able to maintain lubrication without break-down up to a high temperature, it would be difficult to say; as it would be also to state its precise importance in a practical lubricant for other types of machinery. In general, however, a high seizing temperature in this type of test would probably always go with an ability to withstand a high (load × speed) factor without seizure.

A series of tests on this machine of special interest are those

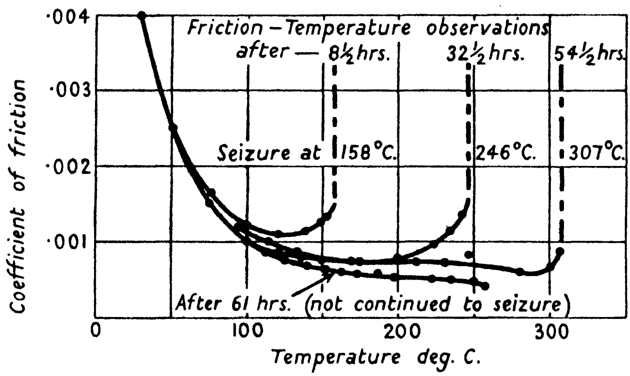

FIG. 51. Rise of seizing temperature and decrease of friction with the progress of oxidation in a mineral oil.

described by King[21] in which it was shown that the extreme limits of seizing temperature and of the minimum coefficient of friction appear to be associated in a mineral oil with the oil being in a partially oxidized condition. Under these conditions the minimum coefficient of friction and the seizing temperature reached the phenomenal figures of 0·00045 and 309° C. The same oil when first tested, before any oxidation had occurred, gave the figures 0·002 and 100° C.

The steady rise of the seizing temperature and fall of μ_{min} after long periods of running under oxidizing conditions are illustrated by the curves in fig. 51. The test conditions throughout were a surface-rubbing speed of 11·3 ft. per sec. and a load on the journal of 1,000 lb. per sq. in. of projected area; which corresponds approximately to 2,000 lb. per. sq. in. over the normal arc of wear.

ART. 34. *Oil consumption.*

In any high-speed internal-combustion engine an amount of oil which is not less than 0·005 pt. per B.H.P. hour is consumed, and the amount may be ten times that figure in an engine which is due

for overhaul. It may be taken that every bit of this consumed oil has by some means found its way up to the pistons into the combustion spaces; for the amount of direct leakage from the crankcase and valve gears can be reduced to a negligible proportion.

Some of the oil left on the cylinder bores may perhaps be completely burnt to carbon dioxide and water, and pass out as such through the exhaust valve, but probably the larger portion is carried out by the exhaust gas in the form of semi-oxidized products and even, when the oil consumption is excessive, in the form of oil molecules and their cracked products.

The problem of reducing oil consumption is therefore purely one of limiting the amount of oil getting up past the pistons to what is necessary for cylinder lubrication; and it is of some interest to arrive at a sort of basic minimum figure, on the assumption, for example, that a film 1 molecule thick is removed by oxidation or evaporation every working stroke. Taking for the basis of one's arithmetic a 12-cylinder engine of bore and stroke 5 in. $\times 5\frac{1}{2}$ in. and yielding 500 h.p. at 2,250 r.p.m., and assuming that the thickness of a monomolecular layer is between 10 and 20×10^{-8} cm. (see table 22), then the volume of oil consumed per hour would be from about $\frac{1}{10}$ to $\frac{1}{5}$ pt., or about $\frac{1}{25}$ of the minimum rate of consumption achieved in practice.

It has already been stated in art. 27 that even under the piston rings there must be a fluid film of appreciable, though widely varying, thickness, so that the film of oil left on the walls during a working stroke is probably on an average several thousands of molecules thick, and we may fairly conclude that when a 500 h.p. engine consumes only $2\frac{1}{2}$ pt. of oil per hour most of the oil consumed is actually burnt or evaporated off the cylinder surfaces, and that there is very little preventable waste.

If an engine can run safely at the minimum figure of 0·005 pt. per B.H.P. hour, then anything above that rate of consumption simply means that more oil than is necessary is passing the pistons, and the excess is being partially burnt or evaporated away; and the reasons why oil consumptions vary so widely and are sometimes ten times the minimum figure must now be considered.

The physical conditions which control the passage of oil up past a piston are still very uncertain. It used to be thought that the movement of oil to the combustion side of the piston was influenced by the gas pressure in the cylinder, and that the flow of the oil was more rapid when an engine was run throttled down because it was then aided by the partial vacuum above the piston during the suction stroke. This idea was proved to be fallacious by Ricardo, with a piston-testing machine in which the oil passing the piston was

collected and measured while various gas pressures, ranging from a 20-in. vacuum to a positive pressure of 45 lb. per sq. in., were maintained above it. Changes of gas pressure were found to have no influence on the amount of oil collected.

The apparent increase of the amount of oil in the combustion space during throttled running is due to an accumulation of that normally passing, because the cycle temperatures are too low for the oil to be burnt and got rid of, as it is at full load. When the throttle is afterwards opened this accumulation of oil can only be very partially burnt, for lack of air, and the consequence is the well-known cloud of blue smoke.

The important factors in oil consumption, apart from piston design and the condition of tune of the engine, are the design and clearances of the connecting-rod big-end, which affect the quantity of oil thrown on to the cylinder walls; the viscosity of the oil at the piston temperature; and the speed. In a given engine, with a given oil, the rate of consumption depends almost entirely on the speed of revolution, and hardly at all upon the throttle position. In so far as the rate of oil consumption has been shown to increase with the load of an engine at constant speed, this is the result of a higher piston temperature which reduces the oil viscosity at the critical points, where it gets past the rings.

An increase of speed has a twofold effect, for besides increasing the rate of passage of the oil past the piston when the supply to the cylinder walls is kept constant, it will also, in practice, lead to a greater quantity of oil being thrown out by the crankshaft on to the cylinder walls. In experiments conducted at constant speed, but with an increasing rate of oil supply, the rate of consumption has been found to increase steadily in proportion to the rate of supply.

Some rational explanation of the process whereby the oil passes from the under to the upper side of a piston will now be attempted.

The practical man knows that within limits, at any rate, the use of a more viscous oil will mean a lower oil consumption; and that by the addition of 'scraper' rings to the piston, or by increasing the radial pressure and reducing the side clearance of existing rings, the same result can be achieved; but how and why the oil gets past the rings at all, at the rate that it does, cannot be explained by a reference to their 'pumping action' as they move up and down in their grooves when the piston motion reverses.

That the passage of the oil past the piston is closely bound up with the behaviour of the rings there is no doubt; but experiments to elucidate the part played by them have led to puzzling and inconsistent results and it is clear that it cannot be dismissed by a

simple reference to their pumping action. It must be remembered, in the first place, that apart from the slight rocking of the piston due to its clearance and the obliquity of the connecting-rod, the motion of the rings and piston is entirely symmetrical, end for end. There is therefore no obvious reason why the rings should pump oil one way more than another, except that fresh supplies of oil are constantly provided from below by the connecting-rod big-end. But this does not suffice to explain the one-way flow, for if one pictures a ring as shifting from top to bottom of its groove on account of the piston reversal while the ring itself remains stationary at the ends of the stroke, any fluid pressure in the oil set up by the piston motion will have dropped to zero as it comes to rest, and there is then little reason why the oil should flow one way more than another just because the supply is more ample on the under side. A rather different sequence of events to explain the upward flow of oil, which does in fact take place, is outlined below.

The aero-engine piston normally carries both 'gas rings' and 'scraper' rings. Fig. 33 on p. 77 showed an elevation and half-section of a typical piston. It has two gas rings *A* and *B*, and two scraper rings *C* and *D*. It is the function of the gas rings to form a gas-tight joint between the piston and cylinder, and they are always disposed in plain, closely spaced grooves near the top of the piston. There may be one or two scraper rings, always below the gas rings, and if there is more than one, the second is often near the bottom of the piston skirt, as at *D*. The scraper-ring grooves are provided with drain holes, *E*, which connect the back of the groove with the under side of the piston, and the rings themselves are commonly cut away as shown in the enlarged cross-section at *F* in fig. 33. Such a form of ring enables a high pressure to be maintained on the cylinder wall and also provides a freer path for the oil scraped from the walls during the down stroke to reach the drain holes. The open groove below the scraper ring *C*, also provided with drain holes, is to collect, and provide the first path of escape for, the excess oil from the cylinder walls.

The important part played by the scraper ring in the matter of oil consumption has been brought out by observations extending over long periods of running. It has been found, for example, that in aero-engines engaged upon a regular commercial service the rate of consumption increased from 5 to 25 pt. per hour between the beginning and the end of a period of 500 hours. After an overhaul, and removal of the sludge from the scraper-ring grooves and drain holes, the rate of consumption was reduced again to the lower figure.

Since the escape of surplus oil through the drain holes is clearly a vital factor, it is a fair conclusion to draw that when they become

blocked a considerable fluid pressure is created below the ring during the down stroke of the piston, as the oil is scraped from the cylinder wall and trapped between it and the piston-skirt, with no free path of escape. When the piston does not carry a scraper ring at the bottom of the skirt, the fluid pressure will build up on the lower side of the lowest ring. Normally this would be a scraper ring, and a large part of the trapped oil escapes through the drain holes to the under side of the piston. Omission of the scraper, or its ineffectiveness through inadequate radial pressure or through an inadequate size of, or blockage of, the drain holes, is always accompanied by a high rate of oil consumption.

Some oil is of course left on the walls by the scraper and subsequent rings, and this can be understood as due to two circumstances. Firstly, no cylinder bore and no piston-ring surface is perfectly cylindrical, so that actual contact between the two must be only at points round the circumference, separated by lengths where there may well be a clearance of 0·001 cm. or less. And secondly, the scraping edge of a ring must be slightly rounded when considered on the scale of hundredths of a millimetre, and this, combined with the unevenness of the surface already referred to, may enable the ring to 'ride-up' on the oil film in the manner of the Michell bearing. Support for this conception is provided by the experience that at very high piston speeds, of the order of 3,000 ft. per min., something in the nature of a critical speed has often been observed at which the rings appear to loose control of the oil and the rate of consumption increases very rapidly.[4] At such a speed one may suppose that the average Michell pressure under the working surface of the ring becomes equal to the radial pressure due to the elastic forces in the metal. The break-down speed can be raised by increasing the elastic pressure of the ring on the cylinder surface.

Further support for the conclusions given in the foregoing paragraph is afforded by the fact that by careful equalization, all round, of the radial pressure exerted by the rings, it was found possible to reduce the normal oil consumption of an engine by amounts varying between 25 and 50 per cent.; and also by the observations of Mougey,[22] confirmed by Ricardo, that although at moderate speeds an increase of viscosity reduces the rate of oil consumption, there may come a point in high-speed engines beyond which this is no longer true. The phenomenon is only noticeable at high speeds. The rate of increase of the consumption with speed of a highly viscous oil becomes much more rapid beyond a certain point, and its consumption as compared with a less viscous one at lower speeds becomes reversed. These are results which can be explained on the

supposition that the high viscosity, while reducing the oil-flow round the back of the rings, enables the rings to ride over, and leave behind, a thicker film on the cylinder wall.

The very slight effect of a rise of cylinder wall temperature upon oil consumption can also be explained along the same lines, for although it may promote, for example, a slightly increased flow of oil behind the rings, owing to a reduced viscosity, this lower viscosity will at the same time have an opposite effect by reducing the oil film between the outer surfaces of the rings and the cylinder wall, and increasing the efficacy of their scraping.

The calculation given earlier showed that the minimum practical consumption of 2½ pt. per hour in a typical 500 h.p. engine corresponded to the removal of a film about 25 molecule diameters thick, or, say, $2{\cdot}5 \times 10^{-6}$ cm. The average film left by the rings due to unevenness of the surfaces may easily be many times thicker than that, so that there is no need to seek for a further explanation in regard to the oil which gets past the rings and is consumed in a well-designed cylinder with efficient scraper rings.

When, however, owing either to the design or to blockage by sludge, the fluid pressure built up below the rings is not adequately relieved, it seems likely that the excessive oil flow past the rings which then occurs is due to a different phenomenon. During the descent of the piston, until the point of maximum velocity is reached at about mid-stroke, the rings must be held against the upper surfaces of their grooves by their own inertia, by surface friction, and by the fluid pressure which is rapidly building up beneath them. Beyond the point of maximum velocity the rings will remain at the tops of their grooves only so long as the friction force, aided by the fluid pressure, is greater than the inertia force on the ring. In a high-speed engine the inertia forces are very large. In the typical engine for which the oil consumption was given earlier the mass of one ring is just under 2 oz. and the inertia force upon it due to the deceleration of the piston is, at the end of the stroke, no less than 50 lb. at a speed of 2,250 r.p.m. It is impossible to say what the fluid pressure in the oil may be, but one may deduce from the mechanical efficiency of the engine that the average friction force between one ring and the cylinder surface cannot be more than 10 lb. at the outside. Direct experiment has shown that, on a piston 5½ in. diam., when there are several rings present, the average friction force per ring is about 6 lb. It is clear, therefore, that long before the end of the down stroke, while the piston velocity is still high and while there is still a big fluid pressure below the rings, the inertia of the ring will cause it to leave its seat against the upper surface of its groove and to move

down to the lower one. While the ring moves from the top to the bottom of its groove there is a free passage open for oil to pass round behind it from its lower to its upper side under the fluid pressure. As the speed of revolution is increased, not only is the average fluid pressure increased thereby, but also the rapidly increasing inertia force upon the ring will cause it to leave its seat earlier on the down stroke than before, when the piston velocity is nearer to its maximum.

This picture of what probably takes place attempts no more than a qualitative explanation, but so far as it goes it does appear to fit all the facts with one, somewhat doubtful, exception. It is consistent with the observed effects of oil viscosity and of the functioning of the scraper ring upon the rate of oil consumption, and it explains the very rapid increase of the consumption with an increase of revolution speed.* What it does not explain is a surprising observation made by Ricardo that a piston fitted with one or more gas rings, but no scraper ring, passed twice as much oil when fitted with four gas rings as it did with only one. These observations, however, seem likely to be contradicted by further investigations with a different apparatus, and it is not possible to say for certain what undetected factors may have been operating to falsify the apparent meaning of the results. One factor of undoubted importance in any attempt to reproduce the working state of a piston in regard to the flow of oil past it, is the exact way in which the oil is collected, for measurement, after it has passed to the upper side of the piston. Obviously the conditions in this respect must be utterly unlike those of practical working where the major part of the oil is burnt, and there is evidence that the measured results obtained have been much influenced by the method adopted for collecting the oil.

The outline given above appears to offer an adequate explanation of how the oil gets from the lower to the upper side of a ring during the down stroke. The reverse process does not happen upon the following up stroke for several reasons. For one thing there is no ample supply of oil present to build up the fluid pressure above the rings. The only oil there is what was left behind after the previous stroke. Equally important are likely to be the facts that the main gas rings are near the top of the piston and that the piston clearance above the rings is normally greater than below them. Instead of having a long and narrow space for the fluid pressure to build up in one has, therefore, a very short and a wider one. This, and the scarcity of oil, are sufficient to explain why the balance of flow is upward past the rings.

* This variation with speed is by no means regular, but has been found to be roughly proportional to between N^2 and N^3, with an increase even more rapid when the critical speed already referred to is approached.

Apart from the broad questions of the number and design of the scraper rings, there are many points in the detailed design of a piston, and its condition, which may profoundly affect the rate of oil consumption. An increase of the side clearance allowed between the rings and their grooves, for example, invariably puts up the rate of oil consumption: a fact which is clearly consistent with the suggested description of how and why the oil gets past the rings; as, also, are the observed effects of speed and of the viscosity of the oil at the working temperature.

In aero-engine pistons the side clearances of the rings are generally graduated from about 0·008 in. in the top groove to 0·003 in. below the second ring; for although the smallest clearance would give the lowest oil consumption it is found that a small clearance where the piston is very hot leads to rapid gumming of the oil in the grooves and to sticking of the rings. A clearance of at least 0·008 in. is necessary in the top groove to allow the first gummy products of the overheating of the oil to be washed away by fresh supplies.

In road-transport engines wear during service is of great importance, both the unequal radial wear of the cylinder and piston, and also in the form of an increase of the side clearance of the rings. Ottaway[23] has given figures relating the percentage increase in the rate of oil consumption between the beginning and end of a running period of 20,000 miles with the wear of the top ring groove per 1,000 miles. His results covered four different types of piston, for which the rates of wear were 0·00012, 0·00025, 0·00027, and 0·00040 in. per 1,000 miles. The corresponding increases in the rates of oil consumption during the period were 21, 41, 53, and 140 per cent. The two were thus roughly proportional, except for the piston showing the most rapid wear, with which the oil control had for some reason deteriorated more rapidly.

The importance of ring groove and other forms of piston wear was further demonstrated by running new pistons and rings in worn cylinders, and vice versa. In the former case the rates of oil consumption were comparable with those in new engines, whereas old pistons in new cylinders continued to show an excessive oil consumption.

For much further information about oil consumption in road-transport engines the reader is referred to the papers by Mougey and Ottaway already mentioned. Much of the data, however, is hardly relevant to the aero-engine, because the amount of wear in the latter during the 500 hours or so of running between overhauls is not so great as to produce effects of an importance equal to those from sludge formation in the ring grooves and draining holes.

VI

THE PRINCIPLES OF HEAT TRANSFERENCE FROM A SOLID TO A FLUID

ART. 35. *Heat transference and surface friction.*

On the fastest aeroplanes the necessary dissipation of the waste heat from the power plant is within sight of placing a limit upon the maximum speed attainable. This follows from the consideration that the rate of heat dissipation increases in proportion to the speed, but the necessary power of the engine in proportion to the $(\text{speed})^3$. It is a fact that it was impossible to run the engines in the Schneider Trophy race of 1931 continuously at full throttle because of inadequate cooling; and this in spite of the fact that a very large proportion of the whole surface of the wings and body was employed as radiator surface, either for the cylinder-cooling water or for the oil.

Whether the heat is carried away from the engine by water, and passed on to the air through a radiator, or whether it is dissipated direct, as from an air-cooled engine, the problem reduces in the end to the fundamental one of increasing in every way possible the rate at which heat can be carried away from a metal surface by an air-stream flowing past it.

In the present chapter the essential principles which control the transfer of heat from a solid to a fluid will be discussed; and in the following two, the application of these principles in practical designs, first in air-cooled cylinders and then in radiators for the liquid-cooled engine.

While flowing past a solid surface, a fluid, whether liquid or gaseous, can only remove heat from it through the movements of its own molecules, which impinge upon the hot surface and acquire more energy in doing so. If we imagine a perfectly stagnant layer of fluid close to the surface, the passage of heat away from the surface, through it, would be only by conduction; and the same would be true of the passage of heat through layers of fluid in steady laminar motion near the surface. Under those conditions heat will only pass from one layer to another, normal to the direction of flow, by reason of molecular movements across the imaginary boundaries of the layers; that is, by conduction. As soon as there is any eddying motion, however, so that there are molar movements towards and away from the surface, heat will be carried by convection away from the hotter layers near the surface. The more turbulent the flow near the surface, the more rapid will be the convection of heat away from

it. Unless some degree of convection were present to assist the mere conductivity of a fluid such as air, the rate of dissipation of heat from a metal surface would be extremely small.

When a fluid like a lubricating oil is flowing past a solid surface it is easy to imagine the flow near the surface to be laminar, and that there is a thin layer immediately in contact with the surface which does not move at all. The tangential velocity v of a layer at a distance z from the surface gradually increases with z, until finally it equals the mean velocity of flow V_m of the fluid past the surface. According to the definition of the coefficient of viscosity μ in the fluid, the tangential force of friction at a surface of separation between two layers at a distance z_1 from the surface will be

$$\mu\left(\frac{dv}{dz}\right)_{z=z_1},$$

and the tangential force on the surface itself will be the value of this expression in the limit when $z = 0$.

The phenomenon of viscosity, which shows itself as a frictional force between the laminae, is due to an exchange of momentum across the boundaries, and in laminar flow must be due to molecular movements only; that is to say, viscosity in laminar flow rests upon the same physical basis, namely molecular movements from layer to layer, as conductivity for heat.

A gas has viscosity, although numerically of an altogether different order from that of an oil, and the conditions of the flow of air near the surface of a solid are generally similar to those with which we are familiar in a viscous liquid. Immediately in contact with the surface is an infinitesimally thin stationary layer; next to this, air in laminar motion of rapidly increasing velocity; and within a small distance the tangential velocity will have risen to the mean velocity of the free air-stream as it sweeps past the surface in more or less turbulent motion. Within the region of laminar flow we are within the 'boundary layer', and here the transference of heat from the surface to the main air-stream is mainly, if not entirely, by conduction.

The velocities and temperature gradients in the air within the boundary layer have of recent years been explored both mathematically and experimentally; but long before these developments, and before the idea of the laminar layer had been fully expounded, Osborne Reynolds[24] had pointed out, in 1874, that whenever a fluid stream moves past a solid surface, it is the motion set up in the fluid near the surface that is the medium, as it were, by which heat can be transferred to the fluid and also by which the momentum of the stream is reduced by skin friction. And he concluded that an intimate

relationship must exist between surface friction and heat transference, each of them depending, as they do, upon the same physical phenomena in the fluid. Later on, Lanchester[25] applied the same idea to the problem of the radiator for an internal combustion engine, and deduced from observations on skin friction the minimum area of heat-dissipating surface required for an engine of a given horse-power.

Although Reynolds had pointed out that 'ultimately it is by conductivity that the heat passes from the walls of the pipe to the fluid', it was left for G. I. Taylor[26] and also, independently, Prandtl to put into mathematical form the complete process of the passage of heat from the surface to the fluid, at first by conduction through the laminar boundary layers, and thence by convection beyond them.

The simple theory of Reynolds and Lanchester starts from the supposition that, because of the identity of the physical phenomena by means of which the velocity and momentum of a fluid stream in a pipe is reduced by surface friction, and by means of which heat is transferred to the stream, it follows that the ratio of the momentum lost by skin friction between two points along the pipe to the total momentum of the fluid between the points considered, must be the same as the ratio of the heat actually conveyed into the fluid stream by convection between the same two points and the total heat which the fluid stream would have received if every part of it had been carried up to the pipe surface and had thus become subject to receiving heat by convection..

The basis of the argument may, perhaps, be demonstrated more forcibly by imagining an open-ended straight pipe full of fluid, which is stationary except for local turbulence. If the pipe, not the fluid, is set in motion parallel to its axis, it will communicate longitudinal motion to the fluid by virtue of surface friction, and of the assumed local turbulence which carries fluid to and from the walls of the tube. If this were such that within a time dt every particle of fluid within a length dx of the pipe would be brought into contact with the surface, then during the interval dt a certain maximum amount of momentum would have been acquired by that portion of the fluid. In the same way, *mutatis mutandis*, we should reach the condition of maximum possible rate of heat transference. The fluid would in the one case have acquired, in time dt, the full longitudinal velocity of the moving tube, and in the other case its temperature.

In order to express the relationship in symbols, let

dp = pressure difference between two points at a distance dx apart, during the motion,

dT = rise of temperature between two points at a distance dx apart during the motion,
T_s = temperature of the pipe wall between two points at a distance dx apart during the motion,
T_m = mean temperature of the fluid stream,
r = radius of the pipe,
V_m = mean velocity of the fluid,
$W = \rho\pi r^2\,dx$ = mass of fluid between the two points dx apart.

Then the equality of the ratios of momentum and heat transfer explained above are expressed by

$$\frac{\pi r^2\,dp dt}{WV_m} = \frac{W\,dT}{W(T_s-T_m)}.$$

Let Q = the heat transferred to the fluid per sec., and F = the surface friction force, each measured per sq. cm. of pipe surface, then

$$Q = \frac{K_p W\,dT}{2\pi r\,dxdt},$$

where K_p is the specific heat of the fluid at constant pressure, and

$$F = \frac{\pi r^2}{2\pi r}\frac{dp}{dx},$$

whence
$$Q = \frac{FK_p}{V_m}(T_s-T_m). \tag{14}$$

For a given velocity of flow, therefore, the rate of heat transfer is proportional to the surface friction and to the difference between the mean temperature of the fluid and that of the tube surface.

It must not be assumed from this result that increased heat transference could be obtained by employing a rough surface so as to get more surface friction. At such a surface the tangential force may not all be communicated to the surface through the agency of viscosity, and the fundamental condition on which the foregoing analysis was based would no longer hold. It is probable that the tangential force on a rough surface would be due largely to differences of pressure between the windward and leeward sides of the small projections from the surface, and apart from the result of some increase of surface area produced by large-scale roughness it has not been found by experiment that anything can be gained in that way.

The intensity of friction, F in equation (14), will itself depend upon V_m. Under many practical conditions it is nearly proportional to ρV_m^2, so that we may write
$$F = C\rho V_m^2, \tag{15}$$

and hence
$$Q = C\rho K_p V_m(T_s-T_m). \tag{16}$$

The rate of heat transference is here seen to be proportional to the density and velocity of the cooling air-stream, and to $(T_s - T_m)$. C is a constant which, for flow through a pipe, could be determined by experiments quite unconnected with heat transfer or temperatures. All that would be needed would be measurements of the drop of pressure down a known length of the pipe through which the fluid was flowing at a known velocity.

The agreement found between theory and experiment in the transfer of heat to air flowing in a tube will be discussed in Chapter VIII. For reasons connected with the boundary layer, to be fully discussed later, the value of the constant C found from measurements of the pressure drop would not in general give values for the rate of heat transfer in accordance with experiment. For air flow the agreement is fair, and since the observed rate of heat transfer is always somewhat greater than that calculated from observations of skin friction, the latter afford a safe minimum figure. With water, on the other hand, for reasons explained below, the agreement is very poor and the discrepancy in the opposite direction.

Apart from the numerical value of the constant, however, the general relationship between skin friction, velocity, and heat transfer has been confirmed by accurate experiments under a variety of conditions. It is found that the skin friction may always be expressed as proportional to some power of V_m, say V_m^n, and that n may vary from 1·5 for laminar flow to 2 for fully turbulent flow; but whatever the value of n, it has always been found that the corresponding rate of heat transfer was proportional to V_m^{n-1}, in accordance with equation (14).

In the foregoing theory, the effect upon heat transference of conductivity acting alone in the laminar surface layers has been neglected. We now know that rapid temperature and velocity changes occur in these layers, where the general turbulence does not penetrate, and that, therefore, the foregoing mathematical argument should be applied only to that part of the fluid which is outside the boundary layer and subject to convective eddies. For the present it will be supposed that the eddy-free layer extends to a definite thickness t from the surface, although this assumption will have to be modified later.

Further, let

U = the velocity, and
T_1 = the temperature, of the fluid at the outer surface of the eddy-free layer,
K = the thermal conductivity of the fluid.

Now if the skin friction between the eddy-free layer and the rest be expressed in the form

$$F = C\rho V_m(V_m - U), \tag{17}$$

then, by combining this with the corresponding form of equation (14), we have

$$Q = C\rho K_p V_m(T_1 - T_m). \tag{18}$$

For conditions within the laminar layer

$$F = \frac{\mu U}{t}$$

and
$$Q = \frac{K}{t}(T_s - T_1);$$

so that
$$Q = (T_s - T_1)\frac{KF}{\mu U}.$$

We have now to obtain Q in terms of $(T_s - T_m)$ and V_m.

Let
$$f = \frac{U}{V_m}.$$

Then it follows that

$$\frac{T_s - T_1}{U} = \frac{T_s - T_m}{V_m}\frac{\mu K_p}{K}\frac{1}{1 + f\left(\frac{\mu K_p}{K} - 1\right)}, \tag{19}$$

and hence that

$$Q = \frac{FK_p}{V_m}\frac{(T_s - T_m)}{\left[1 + f\left(\frac{\mu K_p}{K} - 1\right)\right]}. \tag{20}$$

If $\mu K_p/K = 1$ this expression reduces to the simpler one of equation (14), obtained when the fall of temperature and velocity within the eddy-free layer was neglected.

It happens that for air the value of $\mu K_p/K$ is not far from unity, being 0·88, and therefore, f being less than 1, the term in the square bracket is also, for air, not far from unity, and the rate of heat transference calculated by the simpler formula gives fairly good agreement with experiment. With water, on the other hand, for which $K = 0{\cdot}0014$, $\mu = 0{\cdot}01$, and $K_p = 1$, the value of $\mu K_p/K$ is about 7, and the effect of taking account of the boundary layer is very much to reduce the calculated value of the rate of heat transference.

In this connexion the figures of table 25, given by Stanton from experiments upon the flow of water and air in pipes, are instructive.

It will be seen that, for air, the agreement between theory and experiment is fairly good, more especially when allowance is made

TABLE 25

Observed and calculated figures for the rate of heat transference between a metal pipe and fluids flowing through it.

Diam. of pipe cm.	Mean flow velocity V_m cm. per sec.	Surface friction in dynes per sq. cm	Temp. Difference	Rate of heat transmission* in cal. per sq. cm. per sec. calculated neglecting boundary layer	calculated by the formula of equation (20)	observed
Air—						
4·48	148·0	8·15	20°	0·0266	0·0286	0·030
Water—						
1·39	123·2	50·6	26°	10·8	—	5·36
1·39	69·0	17·1	26°	6·5	—	3·28

* In a subject like that of air cooling, on the borderline between physics and engineering, both C.G.S. and ft.-lb. units must necessarily be introduced. Each system will be used, in what follows, as seems most appropriate to the occasion, and when necessary conversion factors will be given. 1 cal. per sq. cm. = 2·045 C.H.U. per sq. ft.

for the boundary layer according to Taylor's theory. In making the allowance the value of f has been taken as 0·38, as suggested by Taylor on theoretical grounds. Allowance for the boundary layer has not been made for the water observations because of the uncertainty in regard to the value of f. A value of about 0·16 would be required to bring the observed and calculated figures into agreement.

As a likely approximation this is a perfectly reasonable figure, and to that extent it may be said that the theory receives a general confirmation from actual observations. It cannot, however, represent the true state of things accurately in a quantitative sense, for it was based on the erroneous assumption of a definite thickness for the boundary layer and a definitely assignable value for f or U/V_m. In point of fact the thickness of the eddy-free layer may vary from point to point, and in any one position the change from laminar to turbulent flow occurs gradually, not suddenly, so that there is no definite layer to which the velocity U could be assigned. We must now examine rather more closely the real nature of the boundary layer and the changes of velocity therein.

ART. 36. *The nature of the boundary layer.*

The changes of velocity in an air-stream at points close to a surface past which it is flowing have been calculated by Blasius and von Karman for the surfaces of a thin plate placed edgewise to the stream; and they have also been directly measured by Fage and Falkner[27] close to the surfaces of an aerofoil, by means of an arrangement of

minute pitot tubes, down to distances of a few thousandths of an inch from the surface.

The experimental results upon the aerofoil will first be given, and thereafter the equations by which the thickness and velocity of the boundary layer may be calculated for a thin plate with parallel sides. The aerofoil in Fage and Falkner's experiments was of the symmetrical section shown in fig. 52, and when mounted axially in an

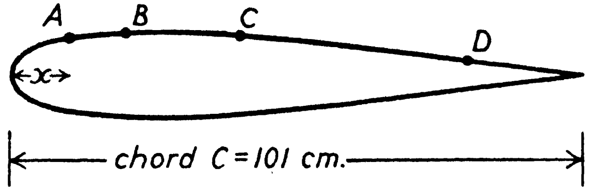

FIG. 52. Symmetrical aerofoil of Fage and Falkner.

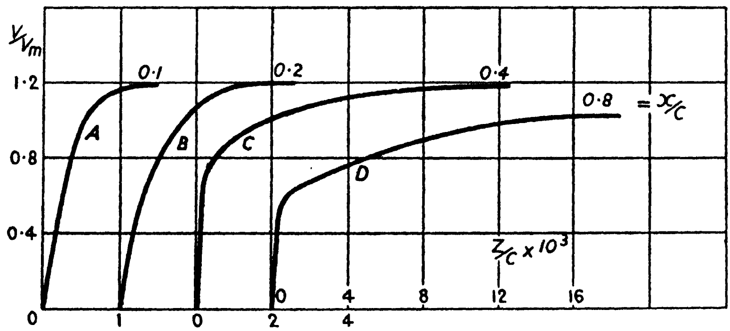

FIG. 53. Air velocities near and parallel to the surface of the aerofoil of fig. 52 when placed axially in an air-stream for which $V_m = 80$ ft. per sec. $c = 101$ cm., Z = distance normal to the surface.

air-stream for which $V_m = 80$ ft. per sec. the velocities parallel and close to the surface were measured at a number of points between the leading and trailing edges.

In order to give a preliminary idea of the nature of the boundary layer and of the variations of velocity within it, there are plotted in fig. 53 the air velocities parallel to the surface (shown as fractions of V_m) at the four points *A*, *B*, *C*, and *D*, situated at 0·1, 0·2, 0·4, and 0·8 of the horizontal distance *c* from the leading to the trailing edge of the aerofoil. The abscissae in fig. 53 are distances normal to the surface, shown as fractions of the chord *c*, which was 101 cm. In order to get the four curves on to the diagram the horizontal scale for curve *C* has been reduced to ½, and that for *D* to ¼ of the scale of curves *A* and *B*.

It will be seen that the four curves reach maximum values of V/V_m at values of Z/c equal to about 0·0015, 0·002, 0·007, and 0·016. In table 26 the corresponding values of Z are given in cm. in column 2,

TABLE 26

Thicknesses of the boundary layer for the aerofoil of fig. 52 for an air speed of 80 ft. per sec.

Position	Full thickness cm.	Values of Z where $\frac{V}{V_m} = 0{\cdot}8$	$\frac{V}{V_m} = 0{\cdot}6$
A	0·15	0·035	0·025
B	0·20	0·05	0·03
C	1·70	0·05	0·016
D	1·6	0·5	0·08

and these represent the full thickness of the boundary layer at the points A, B, C, D under the conditions of the experiment. At these distances from the surface the total head in the air-stream had reached a constant value which was very nearly the same as that in the undisturbed stream. At all except position D the maximum velocity V was about 20 per cent. greater than V_m, but that is a point associated with the shape of the aerofoil which need not concern us here. The important thing to notice is the steady thickening of the boundary layer towards the trailing edge, illustrated in fig. 54 by the curve AB.

Although the curve AB and the figures of column 2 of table 26 may be said to give the full thickness of the boundary layer, it is difficult to estimate its true thickness because of the easy gradient of total head near the outer limit of the layer. In columns 3 and 4 of table 26 are given the values of Z at which the air velocity had reached 0·6 and 0·8 of V_m, and from these figures it is evident that the velocity rises nearly to that of the free air-stream within a small fraction of the full thickness of the boundary layer.

If we were arbitrarily to take the thickness of the boundary layer as the distance at which $V = 0{\cdot}8V_m$, then the measured variation of thickness would be as shown in the curve CD instead of AB in fig. 54. The distance at which $V = 0{\cdot}2V_m$ remained substantially constant for all values of x/c, at about 0·006 cm., and there was strong evidence that within that distance the variation of V with Z was linear.

It will be observed that each of the curves AB and CD in fig. 54 shows a hump, the one beginning at about $0{\cdot}2c$ and the other at $0{\cdot}4c$, where a more rapid thickening of the boundary layer occurs. These humps on the curves indicate a change in the nature of the

flow within the layer itself. It has been shown that although the flow is always laminar for some distance back from the leading edge, there comes a point at which a small-scale turbulence develops in all except the layers nearest to the surface.

Figs. 53 and 54 have now shown up the limitations to the theory

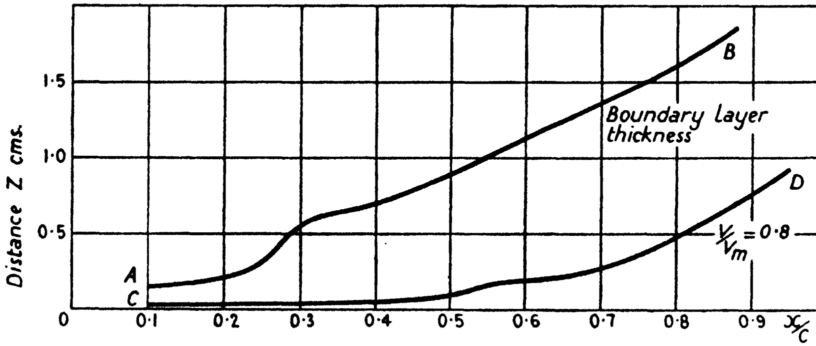

FIG. 54. Thickness of the boundary layer at points on the aerofoil of fig. 52 corresponding to different distances x from the leading edge. Curve AB = full thickness. Curve CD = distances from the surface at which $V = 0{\cdot}8V_m$.

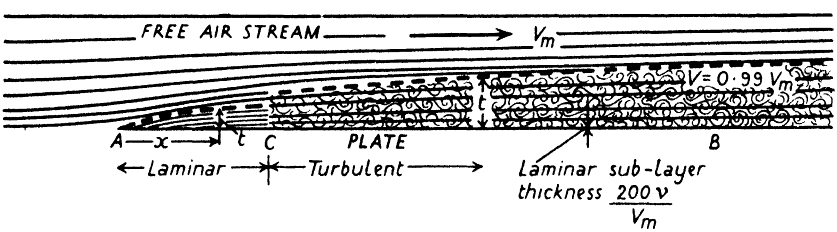

FIG. 55. Shows the changing character of the flow in the boundary layer near the surface of a flat plate.

which led to the formula of equation (20). Not only are there no assignable values of t, U, and f because t is difficult to define and varies from point to point, but also, the degree to which the boundary layer introduces a stage where heat transmission is by pure conductivity will depend upon the existence or otherwise of a transition point to partially turbulent flow within the layer itself.

In the light of what has been said we may now depict the flow of air past one side of a thin flat plate as being somewhat as in fig. 55, where distances normal to the surface of the plate AB are, of course, grossly exaggerated. The discontinuity at the point C, where the nature of the flow changes within the boundary layer, will occur at a distance from the leading edge which depends upon the velocity of the air-stream V_m. The change will always occur at about the same

value of the Reynolds Number ($V_m x/\nu$) in which ν is the kinematic viscosity of the air,* and x is the distance from the leading edge of the plate in the down-stream direction. The critical value of the Reynolds Number, which we may denote by R_x, is from 2 to 3×10^5 for a very thin, smooth plate.

The kinematic viscosity of air is $1{\cdot}59 \times 10^{-4}$ in ft.^{2}/sec. units, and if $V_m = 80$ ft. per sec. this leads to a mean value $x = 15$ cm. as the distance from the leading edge at which the boundary layer will become turbulent. For higher air speeds it will do so sooner, and vice versa. In fig. 54 it will be seen that the hump began to develop on the aerofoil surface, for the same value of V_m, at a point corresponding to $x = 20$ cm.

Within the stretch of purely laminar flow, before the discontinuity, the relationship between the thickness† of the boundary layer t, and the distance x, is given by Blasius as

$$t = 4{\cdot}5\left(\frac{x\nu}{V_m}\right)^{0{\cdot}5}$$

or

$$\frac{t}{x} = 4{\cdot}5(R_x)^{-0{\cdot}5}, \qquad (21)$$

where

$$R_x = \frac{V_m x}{\nu}.$$

For the thickness of the layer beyond the discontinuity a semi-empirical expression of similar form due to von Karman is found to hold good, namely

$$\frac{t}{x} = 0{\cdot}2(R_x)^{-0{\cdot}15}. \qquad (22)$$

This corresponds to the nearly linear relationship $t \propto x^{0{\cdot}85}$, as illustrated in fig. 55. Over this stretch of the surface where the main boundary layer is turbulent there still remains what we may call the laminar sub-layer, of a minute thickness which is probably independent of x and is given by the approximate expression $200\nu/V_m$. At a value of $V_m = 80$ ft. per sec. the thickness, according to this formula, would be 0·005 in.

At 7 in. from the leading edge of a thin plate, where the discontinuity was found to occur when $V_m = 80$ ft. per sec., equation (21) leads to $t = 0{\cdot}06$ in. for the thickness of the laminar layer. With the change to a turbulent layer the thickness rapidly increases to 0·22 in., and thereafter the rate of increase is such that if the surface

* $\nu = \dfrac{\mu}{\rho} = \dfrac{\text{viscosity}}{\text{density}}$.

† Defined as the distance from the surface at which $V = 0{\cdot}99V_m$.

extended far enough the turbulent layer would be 0·3 in. thick at 1 ft., and 2·4 in. at 10 ft. from the leading edge.

ART. 37. *Friction and heat transference in terms of the Reynolds Number.*

Returning now to the correlation between the surface friction and the rate of heat transfer when air flows past a thin plate, the mean frictional force per unit area over the surface will depend upon whether the boundary layer is laminar or turbulent; but here again a simple expression can be derived to give the variation of the frictional force in terms of the Reynolds Number R_x, for the constant $C = F/\rho V_m^2$ in equation (15) can under all circumstances of laminar flow be found from the equation

$$C = \frac{F}{\rho V_m^2} = 0{\cdot}66(R_x)^{-0\cdot5}. \tag{23}$$

Here F represents the ***mean*** force per unit area between the leading edge of the plate and a distance x in the down-stream direction where the Reynolds Number is R_x. After the boundary layer has become turbulent the corresponding expression is

$$C = \frac{F}{\rho V_m^2} = 0{\cdot}019(R_x)^{-0\cdot15}. \tag{24}$$

For a thin plate of width b and length l, over which the flow is wholly laminar, the total rate of heat transfer H, as deduced from theory, is given by the equation

$$\frac{H}{bK(T_s - T_m)} = 1{\cdot}32\left(\frac{V_m l}{\nu}\right)^{0\cdot5}, \tag{25}$$

in which K, as before, represents the conductivity of air.

For such a plate the surface friction coefficient $C = F/\rho V_m^2$ would be given by

$$C = 0{\cdot}66\left(\frac{V_m l}{\nu}\right)^{-0\cdot5} \tag{26}$$

and therefore the heat transfer equation might be written

$$\frac{H}{bK(T_s - T_m)} = 2C\left(\frac{V_m l}{\nu}\right). \tag{27}$$

It must be remembered that H represents the total rate of heat transfer from both sides simultaneously.

In a further set of experiments by Fage and Falkner[28] with a very small thin sheet, of width b and length l, past which the flow was

wholly laminar, the actually measured heat transfer was found to be given very closely by the equation

$$\frac{H}{bK(T_s - T_m)} = 1{\cdot}500\left(\frac{V_m l}{\nu}\right)^{0.5} \tag{28}$$

for four different values of l between 1·27 and 0·33 cm., and for values of V_m between 16 and 32 ft. per sec. The width b = 1·27 cm.

In equation (28) the figure 1·500 compares with the theoretical value of the constant 1·32, given in equation (25). As in Stanton's experiments previously quoted the observed rate of heat transfer exceeds that deduced from theory, but the agreement is rather closer than that found for the flow in pipes and quoted in table 25.

It is possible to obtain comparable expressions for the total rate of heat transfer from a plate of dimensions $b \times l$ under conditions of wholly laminar and wholly turbulent flow in the boundary layer, for if the flow is turbulent equation (16) can be applied, and after introducing the dimensions of the plate so that $Q = H/2bl$ we have

$$\frac{H}{bK_p(T_s - T_m)} = 2C\rho V_m l$$

$$= 2\mu C\left(\frac{V_m l}{\nu}\right)$$

in which the viscosity $\mu = \nu\rho$ and K_p is the specific heat of air at constant pressure.

For the two conditions of laminar and turbulent flow we then have

for laminar flow, $$\frac{H}{bK(T_s - T_m)} = 2C\left(\frac{V_m l}{\nu}\right),$$

for turbulent flow, $$\frac{H}{bK_p(T_s - T_m)} = 2\mu C\left(\frac{V_m l}{\nu}\right).$$

It must be noted, however, in comparing these two expressions, that for both types of flow C depends upon the Reynolds Number being proportional to $(V_m l/\nu)^{-0.5}$ for laminar flow, and to $(V_m l/\nu)^{-0.15}$ for turbulent flow. It will be clear from these expressions that for a given value of V_m the friction per unit area will be more uniform over the plate where turbulent flow has set in, for C in those circumstances will vary less rapidly with l.

To what degree the heat dissipation from a particular plate will follow the laminar or the turbulent law will depend upon the air speed, V_m, and the dimension, l, of the plate in the down-stream direction. As was calculated earlier, with an air speed of 80 ft. per sec. turbulent flow and the corresponding rate of heat transfer would

set in at about 7 in. from the leading edge, provided that the plate was thin and was accurately parallel to the air-stream.

Whether or not $(V_m l/\nu)$ is such as to lead to turbulent flow in the boundary layer, the thickening of the layer from the leading edge backwards occurs under all circumstances, and has an important bearing upon cooling by means of fins; for the velocity of the air-flow near the surface of a fin will be affected at a distance from the surface which will increase the greater the length of the fin in the down-stream direction, and when considering their pitch, or spacing, therefore, it is important not to place the fins so close that the low-velocity boundary layer of one will at any point seriously interfere with that of its neighbour. If the conditions of flow past a cooling fin, due either to its thickness or to its not being parallel to the stream, are such as to induce a more rapid breakdown of the laminar boundary layer than has been calculated above, the result will be a thicker boundary layer at the same value of l, but an extension of the more favourable condition for heat transfer, where the layer is turbulent, over a larger portion of the surface.

Actual measurements of velocity in the boundary layer near the surface of a flat plate, like those described earlier for an aerofoil, will be given presently, and in this connexion some experiments by Marshall[29] are of interest in which air velocities were measured immediately in the rear of a thin sheet 0·06 cm. thick and 4·5 cm. long* placed edgewise to the air-stream. The velocities, expressed as fractions of V_m, are shown in fig. 56, from which it appears that the velocity was reduced below that of the free stream, V_m, over a distance of 0·315 cm. Although these curves give velocities immediately behind, and not at the sides of, the metal sheet, they are of particular interest as showing to what distance the air velocity was affected after passing a thin sheet of the same order of magnitude as the cooling fin of an air-cooled engine.

Allowing for the thickness of the metal sheet, the maximum thickness of the boundary layer at slightly over 4·5 cm. from the leading edge was 0·13 cm. Within that distance the fall of velocity showed the same characteristics as the curves of fig. 53. For the first half of the distance, 0·06 cm., the fall of velocity was only 5 per cent., and at 0·025 cm. from the surface the air velocity was still 0·8 of the free stream value of 1,235 cm. or 40 ft. per sec. Calculation of the boundary layer thickness at 4·5 cm. from the leading edge by equation (21) gives a value 0·105 cm. In that equation t is defined as the point at which $V/V_m = 0{\cdot}99$ and examination of the results illustrated

* The sheet was bent into the form of a hollow cylinder placed with its axis parallel to the air-stream.

in fig. 56 shows a value 0·11 cm. for t so defined. The agreement between theory and experiment is therefore almost exact.

These figures of boundary layer thickness will prove to be of much interest in connexion with experiments upon the rate of heat dissipation by a number of similar fins on an air-cooled cylinder set at various pitches, to be considered later.

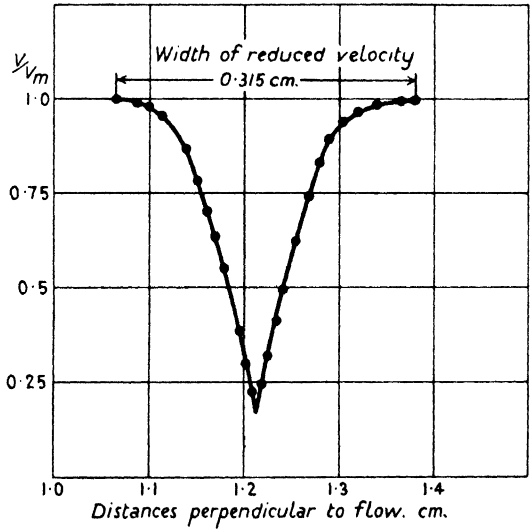

FIG. 56. Variation of velocity across direction of flow, measured close behind trailing edge of a thin plate of thickness 0·06 cm. and 4·5 cm. long in direction of flow.

ART. 38. *Temperature and velocity gradients in the boundary layer.*

The next important step in the problem of heat transference from a solid to a fluid is the measurement of the temperature gradient in the boundary layer, and its correlation with the velocity gradient. Measurements both of temperature and velocity in the air flowing past a heated flat plate were made by Elias,[30] and he showed that the temperature gradient near the surface was in very close agreement with the velocity gradient.

Over the portion of the plate where the flow is laminar the boundary layer is very thin (see fig. 54) and therefore the temperature drops to that of the free air-stream in a very small distance. Over its turbulent portion the boundary layer is thicker, and the full temperature drop is only reached at a greater distance from the surface; but, on the other hand, the effect of the turbulent boundary layer is to carry off the heat more rapidly from the layers very close to the wall. Although the mean temperature gradient in the boundary layer is

less after the layer has become turbulent, nevertheless the gradient very close to the wall is greater, and as a result the rate of heat transfer is increased.

This is illustrated by the two curves in fig. 57 which are typical of the results of Elias's experiments made near the surface of a smooth flat plate. The curves show the rise of temperature in the air near the plate as a fraction of the total temperature difference between the free air and the plate: curve *A* at a point where the boundary layer was still laminar, and curve *B* where it had become turbulent. Each curve represents two series of observations, in one of which the total temperature difference between the air and the plate was twice that in the other, about 18° and 36° C. in the two series. The fact that the points lie on identical curves indicates that as nearly as can be measured the gradient, and therefore the rate of heat transfer, was proportional to the difference of temperature.

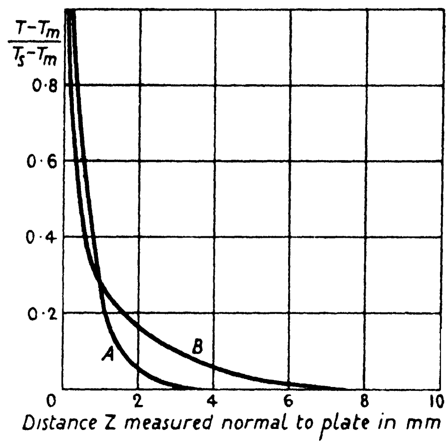

FIG. 57. Temperature gradients in the boundary layer near a flat plate.

The correlation between velocity and temperature in the air is illustrated in fig. 58, where the dots show the air velocity near the surface as a fraction of the free air-stream velocity V_m, and the crosses show the fall of the air temperature below that of the plate, as a fraction of the total fall when the free air-stream is reached. Each curve is plotted against distances from the surface.

We can now visualize fairly completely the two quite distinct conditions under which heat is transferred from the surface of a cooling fin to the free air-stream flowing past it at velocity V_m. Near the leading edge, where the flow is laminar, the heat has to get through a very thin layer, never more than about 0·05 in. thick; but the laminar nature of the flow is bad for heat transfer, since it makes it dependent upon the conductivity of the air. The laminar condition may be expected to extend back for something between 6 and 2 in. from the leading edge on a very thin plate, at air speeds between 60 and 150 m.p.h., but probably for shorter distances on a plate of finite thickness, especially if the free air-stream is itself very turbulent and not exactly parallel to the fin. Beyond the transition point the heat has

to get through a much thicker layer before it is finally carried away in the free air-stream, but the small-scale turbulence in the boundary layer makes the conditions much more favourable, so that the temperature gradient in the laminar sub-layer is very steep, and the rate of heat transfer is greater than when the whole boundary layer is laminar. Under the conditions of a turbulent boundary layer the rate of heat transfer per unit area is nearly proportional to the velocity of the free

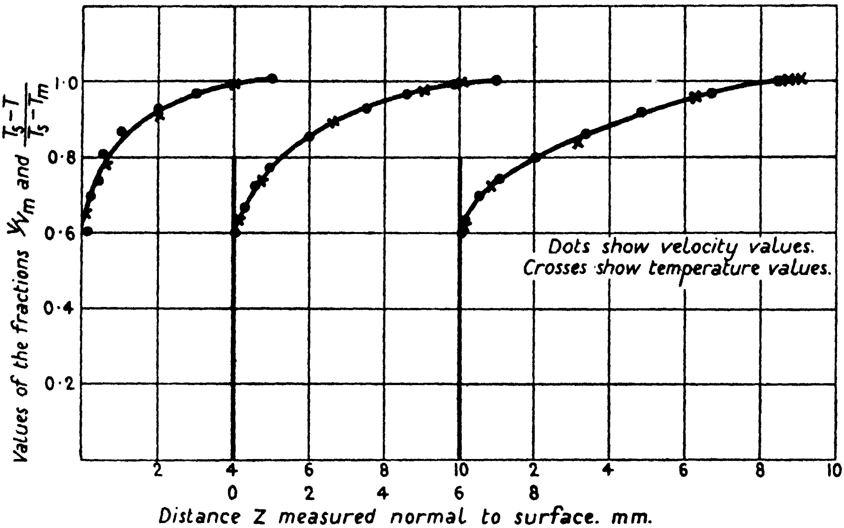

FIG. 58. Correlation between velocity and temperature gradients in the boundary layer.

air-stream and to its temperature difference from the surface. It must be remembered, however, that even if the rate per unit area were strictly proportional to the velocity V_m, the rate of total heat transfer from a plate of given dimensions would not be proportional to V_m; for every change of V_m would alter the position of the transition point, and therefore also the proportion of the plate subject to laminar and to turbulent conditions in the boundary layer.

ART. 39. *Surface temperature and surface roughness.*

It has been stated above that the rate of heat transfer is closely proportional to the temperature difference between the plate and the free air-stream, and over the laminar portion it will also be proportional to the conductivity of the air. It remains to consider to what extent the actual temperature of the air in the boundary layer will influence the rate of heat transfer. The conductivity of air increases with temperature by reason of the increased molecular velocities,

and we might expect, therefore, that the hotter the surface the greater will be the rate of heat transfer per unit of temperature difference above that of the air. This is, in fact, found to be the case.

Stanton[31] made a series of experiments, to be dealt with more fully later on, in which he measured the rate of heat dissipation from a series of electrically heated copper fins, arranged round the barrel of a dummy wooden cylinder of 4·5 in. diameter. The rate was measured for different fin temperatures between 30° C. and 155° C., with air temperatures of 15°–18° C. Stanton found that the rate of heat transfer in C.H.U. per sq. ft. per min. per deg. C. could be expressed by the equation

$$q = 0{\cdot}0057(1+0{\cdot}0075T_m)V_m^{0{\cdot}73}, \tag{29}$$

in which T_m is the arithmetic mean of the fin and air temperatures in degrees absolute, i.e. it is to a first approximation the mean temperature absolute of the boundary layer. V_m is the air velocity in miles per hour.

In order to illustrate the magnitude of the temperature effect it may be pointed out that, with a constant air temperature of 15° C., the raising of the mean fin temperature by 50° C., from 105° C. to 155° C., would increase T_m from 333 to 358 and the temperature factor from 3·50 to 3·68. The rate of heat transfer per degree of temperature difference would therefore be increased by 5 per cent. on a set of fins of the type used by Stanton. They were 114 mm. inside and 146 mm. outside diameter, 0·55 mm. thick, and spaced 8 mm. apart.

The question of the variation of the rate of heat transfer with air velocity, for circular fins of this type, will be deferred until the heat dissipation from complete cylinders is considered.

It is appropriate at this point to say something of surface roughness. Experiments by Stanton[32] upon the flow of water in pipes have shown that a substantial increase in the rate of heat transfer is produced by roughening the surface, and that the increase in the rate of heat transfer is closely proportional to the increase in the surface friction. Further experiments by Stanton, in which the heat was dissipated to an air-stream from cooling fins on a circular cylinder, failed to show the smallest difference between the conditions when the fins were smooth and when the surfaces were corrugated into serrations 0·5 mm. deep and 1·8 mm. pitch.

The experiments of Elias were made both on smooth and rough plates, and there was some indication that the transition to a turbulent boundary layer occurred at a lower Reynolds Number near the rough than near the smooth plate. The difference, however, was not

large. Even if a roughened surface did promote a readier change to a turbulent boundary layer, it would not necessarily follow that the rate of heat transfer would be increased; there might be a compensating effect through the roughness having increased the mean thickness of the laminar sub-layer, which would still cling to the hill-tops and fill up the hollows, like mist in a mountainous country.

A further practical point which was tested by Stanton was whether rapid vibration, such as occurs with the cylinders of an air-cooled engine, has any effect upon the rate of heat transfer through a power of setting up a small-scale air turbulence near the surface. The results of the experiments, however, were entirely negative.

VII

THE AIR-COOLING OF CYLINDERS

ART. 40. *A statement of the problem.*

Of the heat generated per cycle in the cylinder of a high-speed engine, 25–30 per cent. can be converted into useful work on the pistons, 45–50 per cent. is rejected with the exhaust gases, and 20–25 per cent. has to be got rid of, either by direct transference to the surrounding air in an air-cooled engine, or, in a water-cooled engine, partly by direct cooling and partly through the intermediary of the cooling water, which later dissipates the heat to the air in a radiator. Of the 20–25 per cent. which is dissipated to the surrounding air, not more than 15–20 per cent. would be conducted through the walls of the cylinder; the remainder would be removed from the piston and bearings by the lubricating oil, and later dissipated either from the crankcase or from a special radiator.

Gibson[33] has given figures for two air-cooled aero-engine cylinders, designed in 1916, from which the total heat dissipated was estimated to be 110 per cent. of the B.H.P., and of this rather more than half was dissipated directly to the cooling air-stream. Heron[34] gives 60 per cent. of the B.H.P. as the heat to be dissipated directly by the external cooling surfaces of an air-cooled cylinder; and we may accept this figure, with a proviso that some reduction, possibly to 50 per cent., is to be expected among high-speed engines of high compression ratio, and especially if they are operating with a rich fuel-air mixture.

The rate of conduction of heat through the walls of the cylinder and combustion space must be sufficient to keep the inner surface temperatures below certain practical limits, and there are two points of view from which these limits must be considered. The first concerns the cylinder barrel, over which lubrication must be maintained; the second, the much higher range of temperatures at the surface of the combustion space, and more particularly at certain critical points such as near the exhaust-valve seat.

It is really astonishing that a piston can be maintained, at an average speed of more than 2,000 ft. per min., in rubbing contact with a cylinder wall which is swept by gases at a white heat for more than half its working life. The surface of the lubricating oil is badly scorched during every working stroke, but the film is continually renewed, and in this way lubrication can be maintained even under these severe conditions, provided the metal of the cylinder walls is

of the fin below its root temperature, and the observed efficiency should correspond to the value calculated from theory. When the boundary layers begin to interfere, however, the assumed conditions of the theory are no longer fulfilled. In these circumstances the observed efficiency falls away in spite of a *rise* in the fin temperature to nearer the root value, because the air-stream has become ineffective in carrying away the heat.

At a pitch of ½ in. there is little or no chance of one fin interfering with the air-flow past its next neighbour, and the condition of a single fin subjected to the full air-stream velocity V_m, postulated by theory, must be closely fulfilled. A comparison between the observed and theoretical efficiencies, taking the experiments at ½-in. pitch, although showing good agreement at $V_m = 87$ m.p.h. (0·79 from theory against 0·75 observed) reveals the discrepancy that according to theory the efficiency should be lower at the higher speed, owing to the greater emissivity q; whereas in the experiments for the widely spaced fins, the efficiency was higher at 158 than at 87 m.p.h. The discrepancy is explained if the average emissivity per unit area over the fins at the higher air-speed was in fact greater than had been found by experiments on the plane plate, and used in column 5 of table 30. Such a thing is highly probable, because the mean emissivity will depend upon how much of the surface is subject to turbulent flow in the boundary layer, and at the very high air speed it is particularly likely that turbulence would set in earlier at the sides of a fin of finite thickness than over the surface of a flat baseplate fitted with a properly streamlined edge leading on to it. If there was this increase of the turbulent area over the fins it would mean that the denominator used in finding the experimental efficiency (column 5 of table 30) was taken too small, and too high a value of the fin efficiency is the result.

Comparisons of theory and experiment in the matter of the fin efficiency for an isolated fin, however, though of interest, are not of much practical importance. What *is* important is that fig. 61 provides a clear indication of the minimum fin spacing for a high fin efficiency. In this figure the values of the efficiency found from the experiments have been plotted against the width of clear space between the fins (i.e. the pitch less the fin thickness). If it were not for the interference of the flow past any one fin due to its neighbours on either side, the efficiency would be independent of the pitch and the curves *BA* and *DC* would remain horizontal. Instead of that there was a sudden falling off in efficiency when the space between the fins fell below about 0·15 in.

Reverting now to the figures in table 28, it is clear that some interference of the boundary layers of neighbouring fins would occur at

their trailing edges when the clear space between them was reduced below $2 \times 0{\cdot}17$ in. But the interfering layers would be turbulent, and the forward velocity of the air under those conditions does not fall below $0{\cdot}9V_m$ until the distance from the surface is less than half the full thickness of the layer. We may argue, therefore, that the effectiveness of the air-stream would be still 90 per cent. of its full value when the space between the fins was reduced to $0{\cdot}17$ in. Soon after that its effectiveness should fall off rapidly, firstly because the maximum velocity of the air between the fins will fall away, and even

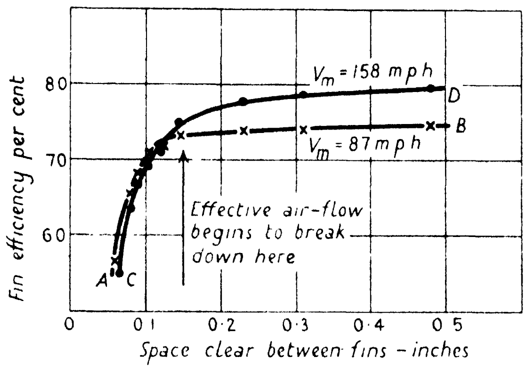

FIG. 61. Variation in efficiency of copper fins at different spacings.

more because very soon a condition will be reached when the *laminar* boundary layers will coincide. This would happen, for $V_m = 87$ m.p.h., when the clear space was reduced to $0{\cdot}08$ in., according to table 28. At this value, or somewhat above it, fig. 61 exhibits a rapid falling away of the fin efficiency. When the fins are close enough for the laminar layers to coincide at the point of transition to turbulent flow it seems possible that the normal development of eddies beyond that point would be damped out, and the rate of cooling be thereby reduced to that corresponding with laminar flow throughout the whole fin length. If this is correct it would explain the collapse of the fin efficiency exhibited in fig. 61.

Even before the fins are close enough for the laminar layers to coincide at the transition point, the interference of the layers beyond that point may, perhaps, have the effect of beginning to damp out the normal eddy formation, for the normal thickening of the layer must be prevented, and this, combined with the expansion of the air by reason of its rising temperature, may cause the flow to take on the characteristics of flow in a convergent instead of in a divergent channel.

Arguing from the data of table 28, therefore, we should expect

the turbulent flow between the fins, necessary for a high rate of cooling, to begin being damped out at a distance between the fins of 0·26 in. for $V_m = 87$ m.p.h. and at 0·14 in. for 158 m.p.h. In the first case the damping out would only begin 1·5 in. from the trailing edge, whereas at 158 m.p.h. it would extend over more than half the length of the fin. Considering that the calculations of boundary-layer thickness only apply strictly to very thin plates, the observed value of 0·15 in. for the clear space at which the normal heat

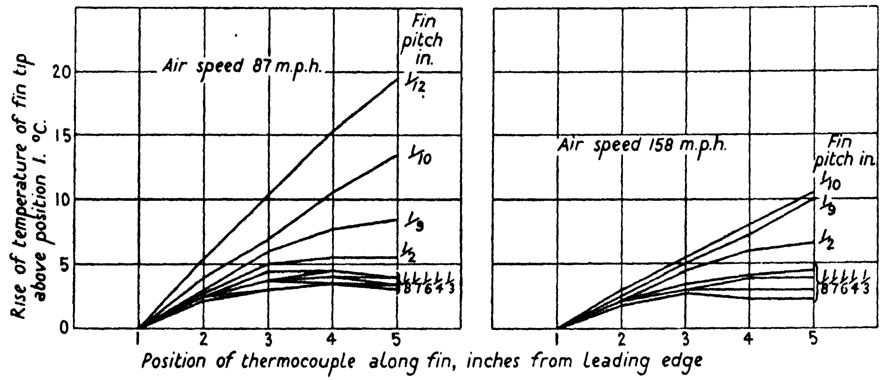

FIG. 62. Rise of temperature along the outer edge of a fin in the down-stream direction.

dissipation of the fins begins to break down must be held to agree very satisfactorily with the data derived from boundary-layer theory.

In flowing past the barrel fins of an air-cooled cylinder the air has to follow a curved path, which will complicate the problem in two ways. The effective length of the fin is uncertain, in the first place, and also the thickness of the boundary layer at the fin surface will probably be affected by the change of direction of the air-stream as it flows round the cylinder at the roots of the fins. Further discussion of the effectiveness of fins round circular cylinders, however, will be deferred for the present, until the results of experiments on the cooling of a complete cylinder are discussed.

With the same test specimen composed of 6-in. straight copper fins, an interesting series of temperature measurements was made at the fin tip, for points at various distances along the length of the central fin of the test specimen. Although actual fin temperatures are not given in the report, the amount by which the fin tip temperature exceeded the air temperature at distances of 1, 2, 3, 4, and 5 in. from the leading edge are given. The results are summarized in fig. 62. There was always some rise of temperature along the fin, as would be expected by reason of the heating up of the cooling air,

but apart from some discrepancy among the observations at ½-in. pitch, the rise was never more than 3° or 4° unless the pitch was less than ⅛ in. For pitch values of ⅛ in. and over, the temperature rise above the first observation point reached its maximum value of about 4° at the centre of the fin length, and thereafter remained nearly constant.

At a pitch of ⅛ in., corresponding to a clear space between the fins of 0·105 in., the critical condition was reached, so that further reductions of the clear space to 0·09, 0·08, and 0·06 in. produced rises of temperature between points 4 in. apart of 8°, 14°, and 19° C.

To sum up, we may picture the rate of heat dissipation from a finned surface as being inevitably diminished by the presence of the boundary layer, and more especially when the flow within it is laminar. Whether this is so or not, there will always, beyond the outer limit of the boundary layer, be some degree of turbulence, even in the generally streamline flow of a wind tunnel; and the presence of the free air-stream at the full average velocity V_m is essential to carry away the heat from the outer layers of a boundary layer in laminar flow.

When air is constrained to flow in the narrow space between two fins, the stream of free air at the full velocity will cease to operate as soon as interference between the boundary layers of two neighbouring fins occurs at the down-stream end. If conditions are such that the boundary layers have become turbulent before they reach the point of interference, then no rapid falling off in the rate of heat transfer is to be expected until the fins are so close that interference occurs near the transition point from laminar to turbulent flow.

Art. 43. *Heat dissipation from finned cylinders.*

Having examined the fundamental conditions which influence the rate of heat dissipation from a finned surface, it remains to consider the results of experiments upon complete cylinders, and so far as possible to reduce them to data of general applicability. Although the results of some excellent researches are available, their rational interpretation is a matter of much difficulty on account of the lack of symmetry in the form of the cylinders, and the wide variations of temperature between different points on them. Take, for example, some of Gibson's results already referred to. He made observations on three different cylinders: two aluminium and one cast iron. From one of the aluminium cylinders, with fins originally of 9 mm. pitch, he later removed, first, every alternate fin so that the pitch was increased to 18 mm., and then all the fins, so that the bare aluminium cylinder and steel liner were left without fins of any kind to assist the cooling.

These are just the kind of experiments needed to link up with laboratory results, but the data summarized in table 31 will show the

TABLE 31

Mean temperatures and mean rates of heat dissipation from three air-cooled cylinders. Aluminium cylinders: 1,600 r.p.m. Cast iron cylinder: 1,800 r.p.m.

Cylinder type and size	*C.R.*	*Fin pitch*	*Air speed V_m m.p.h.*	*Total cooling area sq. ft.*	*Mean temperature above air deg. C.*	*Rate of heat dissipation* C.H.U./sq. ft. per min. deg. C.	*Rate of heat dissipation* Cal./sq. cm. per sec. deg. C.
Aluminium—							
(*a*) 100 × 140 mm.	4·6	12 mm.	64	6·14	105°	0·67	0·0054
(*b*) ,,	,,	,,	60	,,	65°	0·30	0·0024
Aluminium—							
(*a*) 114 × 140 mm.	4·6	9 mm.	63	8·2	108°	0·46	0·0037
(*b*) ,,	,,	,,	60	,,	66°	0·23	0·00185
(*c*) Alternate fins removed	,,	18 mm.	63	4·9	150°	0·66	0·0053
,, ,,	,,	,,	78	4·0	128°	0·79	0·0064
(*d*) All fins removed	,,	Bare cylinder	63	1·8	250°	*	*
,, ,,	,,	,,	78	,,	228°	1·08	0·0088
Cast iron—							
(*a*) 100 × 140 mm.	4·2	8 mm.	63	4·7	146°	0·55	0·0044
(*b*) ,,	,,	,,	60	,,	52°	0·28	0·0022

* Cylinder failed by overheating. (*a*), (*c*), and (*d*) cylinders under power at full throttle. (*b*) Cylinders heated by circulating hot water.

difficulty of doing so. Besides making measurements of the cylinder temperatures and of the total heat dissipated per minute while the engines were running under power, some further observations were made when the heat to be dissipated was supplied by hot water circulated through the cylinders. In table 31 the observations made under those conditions are on the lines marked (*b*). The experiments with every alternate fin, and with all fins removed, are labelled (*c*) and (*d*) respectively.

At first sight the figures in this table appear to show a steady increase of the mean emissivity, i.e. of the mean rate of heat dissipation per sq. ft. per deg. C., as the cooling area was reduced, and as the cylinder temperature became higher. The mean emissivities found from the water experiments were in each case just about one-half of those when running under power. It was pointed out earlier that under comparable conditions the true emissivity of a surface does increase with the temperature; but the increase is quite inadequate to explain the figures of table 31. In Stanton's experiments, the results of which were expressed in equation (29) on p. 155, it amounted to 5 per cent. for an increase of fin temperature of 50° C.

Apart from any increase of the true emissivity, there are two reasons which explain the apparent anomaly.

(1) Of the total heat estimated as being lost from the gases while in the cylinder, only a portion would be dissipated directly by the cooling surfaces. The rest would be carried off by the lubricating oil or conducted away to the crankcase and foundations.

(2) All the cylinders used by Gibson carried a large and heavily

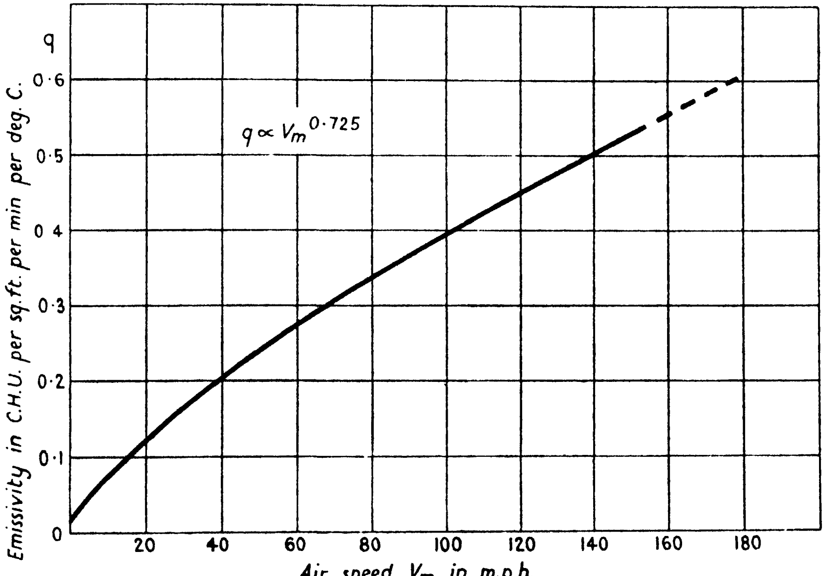

FIG. 63. Mean emissivity for flat surface subject to tangential air-stream.

finned exhaust-valve housing, protruding from the top. The temperatures of these were not measured and included with the other temperatures in arriving at the mean temperature of the cylinder above that of the cooling air. There must have been a very rapid heat loss from these housings when running under power, which would have the effect of giving an unduly high value to the average emissivity.

The figure for the mean emissivity derived from experiments of this type with complete cylinders, in which the rate of total heat dissipation is divided by the total area of the fin surfaces and by a figure representing the mean temperature difference above that of the air-stream, bears little direct relation to the true emissivity q used earlier in connexion with the flow of air past a flat surface. A curve of the mean emissivity for a flat surface 6 in. square at about 65° C., subject to a tangential air-stream of different velocities at 20° C., is given for comparison in fig. 63.

When a cylinder is air-cooled it is subject to an air-stream which impinges at all angles between 0° and 90°, and the rate of heat transfer will vary from point to point accordingly. Only where the flow is truly tangential will the emissivity have the values given in fig. 63, and even where it is tangential the velocities over the surface may bear little relation to the velocity in the free air-stream. The velocity in Gibson's experiments was measured at a point 6 in. in front of the cylinder axis, and he states that a velocity of 64 m.p.h. measured at that point gave the same mean velocity over the whole cylinder surface as 58 m.p.h. measured in the free air-stream. When placed in such an air-stream the mean velocity over the cooling surfaces, as found from pitot tube measurements, was only 36 m.p.h.

During the experiments with the cylinders artificially heated the surface temperatures would be more uniform, and the figures of 0·23, 0·28, and 0·30 C.H.U. per sq. ft. per min. given in table 31 for the three cylinders when subject to an air-stream of 60 m.p.h. are no doubt properly characteristic figures for each of the cylinders in regard to heat dissipation at that air speed. It happens that they are in fair agreement with the figure 0·275 at the same air speed in fig. 63; but this is merely because the direct impact of the air on some parts of the cylinders compensates for a lower mean air speed over the cooling surfaces. It must be remembered that according to Gibson's pitot tube measurements the mean air speed would have been only a little more than 36 m.p.h.

The figures 0·23, 0·28, and 0·30 C.H.U. per sq. ft. per min. could safely be applied to find the approximate rate of heat dissipation from the three cylinders under working conditions, provided the fin temperatures were measured at a sufficient number of places to arrive at a true mean value for the temperature difference. It will be understood, however, that these figures are each characteristic of a particular cylinder design, and depend not only upon the height and spacing of the fins but also, for example, upon the form of the exhaust-valve housing and its position in relation to the air-stream. It has also been stated by Gibson[33] and by Heron[34] that the rate of heat dissipation from an engine cylinder depends upon the material: that a steel surface under comparable conditions is from 5 to 10 per cent. better than an aluminium one, and that bronze is notably inferior to either. Gibson has also stated that the dissipation from an aluminium cylinder was increased 10 per cent. by a coat of glossy black stoving enamel.

Although small differences are to be expected on account of the liability of some metals to form a poorly conducting film of oxide on the surface, we have seen that a roughening of the surface has no

appreciable effect upon its emissivity, and it does not seem that differences of emissivity are likely to be met with of sufficient importance to affect the choice of materials.

As regards the effect of a change of velocity, Gibson found that the rate of heat dissipation from his cylinders was proportional to $V_m^{0\cdot73}$. Experiments upon the test specimen with straight fins, 6 in. long, summarized earlier, showed a variation with velocity proportional to $V_m^{0\cdot747}$, and upon the plane plate without fins, proportional to $V_m^{0\cdot725}$. Gibson's figure of $V_m^{0\cdot73}$ agrees exactly with Stanton's, given in equation (29), which was obtained on a test piece with circumferential fins only, of about the same diameter; and if we accept a variation proportional to $V_m^{0\cdot73}$ as representative of groups of fins on a typical cylinder we shall not be far out. Here we are on more solid ground, moreover, for the result can be related to the basic conditions for heat transfer considered earlier.

It will remembered that for laminar flow past a surface the rate of heat transfer was nearly proportional to $V_m^{0\cdot5}$, and for turbulent flow to V_m. For any set of fins of the order of magnitude of the 6-in. test piece and of an aero-cylinder, subject to air velocities between 50 and 150 m.p.h., we know that the proportion of the surface covered by the laminar layer will vary between $\frac{1}{4}$ and $\frac{3}{4}$ of the whole, and we may regard the experimental relationship

$$H \propto V_m^{0\cdot73}$$

as illustrating the compromise brought about by some of the surface being subject to laminar and some to turbulent flow conditions in the boundary layer.

Turning now to the practical operating conditions in a high-power air-cooled cylinder, 20 B.H.P. per litre of swept volume has been given as a very good output for the larger class of cylinders of 5–6 in. bore. It corresponds to a B.M.E.P. of 130 lb. per sq. in. at 2,000 r.p.m. If 0·6 of the B.H.P. must be dissipated from the cooling surfaces, this means that the total rate of heat transfer must be 12 h.p., or 282 C.H.U. per min., per litre of swept volume.*

As regards the fin area to be provided to effect this rate of heat dissipation, it is unsafe to generalize. For the larger sizes of aero-engine cylinder, of about 200 cu. in. capacity, it was authoritatively stated in 1920 that the limit of external cooling surface which could be crowded on, without exceeding the proper limits of weight and bulk, lay between 0·21 and 0·25 sq. ft. per B.H.P.; and at about the same date Gibson and Heron gave 0·30 to 0·35 sq. ft. per B.H.P. as being possible with small cylinders of about 70 cu. in. capacity.

* Equivalent to 19·5 h.p. or 460 C.H.U. per min. per 100 cu. in.

Heron further gave it as a rough generalization that cylinders of this small size could be adequately cooled when under full power by an air-stream of 60 m.p.h., but that the larger range, of about 200 cu. in., required a speed of 90 m.p.h.

Since that date, increases of r.p.m. and improvement of cylinder design and materials have modified the conditions to a degree which can best be grasped by studying the typical figures of table 32. In

TABLE 32

Cylinder	*Swept volume cu. in.*	*B.H.P. per cylinder*	*Heat to be dissipated 0·6 × B.H.P. C.H.U./min.*	*Internal surface area sq. ft.*	*External cooling area sq. ft.*	*Total cooling surface*		*Heat dissipated per sq. ft. per min.*	*Mean temperature difference*	
						Per B.H.P.	*Per sq. ft. internal*		*At 60 m.p.h. h = 0·30*	*At 100 m.p.h. h = 0·44*
4D (1916), 3.94 in. × 5·51 in., C.R. 4·6, 2,000 r.p.m.	67	16·1	228	0·69	6·14	0·38	8·9	37	123° C.	85° C.
Jupiter, 5¾ in. × 7½ in., C.R. 5·3, 2,000 r.p.m.	195	53*	750	1·33	8·71	0·165	6·6	86	287° C.	195° C.
Rapier, 3½ in. × 3½ in., C.R. 6·0, 3,500 r.p.m.	34	20·9*	296	0·355	3·15	0·151	8·9	94	313° C.	214° C.

* Both these engines normally carry a supercharger. The figures given are the gross B.H.P. per cylinder, including the power for driving the supercharger.

this table the data for one of Gibson's cylinders, of 1916, is given for comparison, and beneath it the corresponding figures for a large and a small cylinder of modern design.† The modern cylinders probably represent the extreme limits in point of cylinder capacity, the larger being of nearly six times the swept volume of the smaller, and this makes the comparison of the cooling data provided by them of all the more interest. The speeds of the two engines provide an equally striking contrast. The short stroke and light moving parts of the Rapier permit a revolution speed which is impossible for the larger engine, and afford a correspondingly high power output per unit of cylinder volume.

The ratio of the external cooling area to the maximum internal surface of the cylinders is not very different in the three designs, and it is interesting that, in spite of the wide diversity both of speed and size between the modern engines, the area of cooling surface per B.H.P. comes out so closely the same in each. The figures of 0·165 and 0·151 sq. ft. per B.H.P. are just about half the figure given by

† From data kindly supplied by Messrs. A. H. R. Fedden and F. B. Halford.

Gibson and Heron in 1922 as applicable to small cylinders. On the assumption that in each case the total rate of heat dissipation is equal to 0·6 of the B.H.P. one arrives at 86 and 94 C.H.U. per sq. ft. per min. as the necessary rates of heat dissipation per unit area, i.e. a little over twice the figure for the cylinder of 1916.

It must be remembered, of course, that while the earlier cylinder was adequately cooled in a wind of 64 m.p.h. the modern engines would require one of about 100 m.p.h. when operating continually at full load. This increase of air speed would account for an increase in the rate of heat dissipation in the ratio

$$\left(\frac{100}{64}\right)^{0\cdot73} = 1\cdot38.$$

It is clear, therefore, that, even allowing for some increase of the mean temperature in the modern over the old cylinder, there is evidence of a substantial increase in the rate of heat dissipation per sq. ft. due to improved design, for this, with perhaps some help from a higher mean temperature, must account for the difference between the 100 per cent. increase of heat dissipation per sq. ft. on the modern cylinders and the 38 per cent. provided by the higher air speed. Although the ratio of external to internal surface shows no increase, and in one case a decrease, improved design both of the fins and of the form of the cylinder head must have enabled a higher average air speed to be maintained over the cooling surfaces for the same velocity in the free air-stream.

The calculated figures for the mean temperature difference given in the last two columns of table 32 have no practical significance, because the real figures for the mean emissivity of the two modern cylinders are not known. The figures used for calculating the mean temperature differences are the highest value of the mean emissivity found by Gibson at $V_m = 60$ m.p.h. when using a uniformly heated cylinder, namely, 0·30 C.H.U. per sq. ft. per min. per deg. C.; and the same figure increased, for the higher wind speed, in proportion to $V_m^{0\cdot73}$. The temperature figures are included only to illustrate the effect of a change of air speed.

The true value for the mean emissivity of the modern cylinders could only be found by taking a sufficiently large number of measurements of the fin-surface temperatures to arrive at a true mean value. The result would certainly be a lower value of the mean temperature difference than that calculated and given in table 32, showing that the improved cylinder-head and fin design had resulted in a higher mean emissivity than that found for the cylinder of 1916 under comparable conditions.

Recent figures for the temperatures and power output obtained from a small supercharged cylinder of a type generally similar to the Rapier, but with improved cylinder-head design, indicate that a mean overall emissivity of about 0·61 C.H.U./sq. ft./min./deg. C. was obtained at $V_m = 125$ m.p.h., which corresponds to 0·36 at $V_m = 60$ m.p.h. Although these tests were under full load conditions this figure 0·36 should probably be compared rather with that of 0·30 in Gibson's experiments than with his higher figures which did not take account of the temperatures of the exhaust-valve housings. In the recent experiments the temperature at the hottest point, under the exhaust-valve seat, was 242° C., and the mean temperature difference for the whole cylinder cannot have been more than 200° C.

The improvement of the mean emissivity from 0·30 to 0·36 at the same air speed represents an improvement of cylinder-head and fin design in the direction of putting the fins where they are most needed, and of providing a flow of air over them which is obstructed as little as possible by the various ports, plugs, and bosses that complicate the design so much.

No figure which might be given for the 'mean' temperature of a cylinder has very much practical significance, where the range of temperature is so wide and when certain comparatively local high temperatures are what really matter. An attempt will be made presently to separate the heat dissipation from the cylinder head and from the barrel, and in that connexion a figure for the mean temperature of each above that of the cooling air will be further considered. The mean temperatures of the head and of the barrel, considered separately, are figures of some practical significance; and the relation between them is important from the bearing it has upon the problem of distortion.

From this point of view, also, the temperature difference from point to point within the head and the barrel are of much importance. A cylinder barrel of 6-in. bore has been found to be 0·012 in. out of round under working conditions, owing to uneven cooling.[4] Coupled with the two limiting temperatures of about 250° C. for the head and 180° C. for the barrel, there should be a maximum difference between the front and back of the steel barrel of about 100° C., with a mean difference of 70° C. to 80° C. Between the top and bottom points of the piston travel there should not be a temperature difference of more than about 30° C. on the rear, or hotter, side—from 180° at the combustion end to about 150° near the crankcase. As regards the cylinder head, it is evident that for uniformity of temperature the cooling air must be directed against that side where

the exhaust valves are situated. It is true that in the early days of the aluminium cylinder-head some designers were misled into believing that the high conductivity of the material would render it unimportant to provide direct air cooling of all the vital parts of the head. Experience has shown that a high conductivity is not enough, and that for an air-cooled cylinder to maintain an output comparable with what is possible in a good water-cooled one, it is essential so to design the form and finning of the head that all the vital points are directly cooled.

Heron[34] has given some results of experiments upon a 4-valve air-cooled cylinder of bore and stroke 5½ in. by 6 in., developing 36 B.H.P., in which the extremes of temperature obtained upon five thermocouples at various points on the head, which were 77° C. and 300° C. when the air blast of 70–75 m.p.h. was directed against the inlet valve side, were levelled up to 112° C. and 190° C. when the cylinder was turned through 180°.

An essential point in this and other successful designs is that the two exhaust-valve housings are separated by a space containing straight fins in the fore-and-aft direction. Between these fins the cooling air has an uninterrupted flow for carrying off the heat from this vital spot, at the centre of the hot side of the head.

It has been mentioned already that the rate of heat dissipation per unit area of surface is much greater for a directly impingeing air-stream than for a tangential one of the same velocity. If the air impinges directly, however, its velocity is largely destroyed, and so also its effectiveness for cooling more than the particular point of impact. The art of good cylinder-head design, therefore, lies in providing for an unimpeded flow of air past the maximum possible area of surface, with some degree of direct impact at the hottest points.

Increased heat dissipation can sometimes be obtained from the barrel fins of an engine cylinder if the cooling air-stream be directed other than at right angles to the cylinder axis. In these circumstances, instead of the fins themselves being almost entirely tangential to the air-stream, some portion of their area becomes subject to the direct impact of the air at whatever is the angle of inclination of the cylinder.

In some recent American experiments[39] it has been shown that the greatest rate of heat dissipation was obtained from a cylindrical test specimen when its axis was inclined at 45° to the air-stream direction, and that the rate was then about 50 per cent. greater than when it was in the normal position at right angles to the stream. The test specimen in these experiments was of 4½-in. internal diameter, while the fins were of height 0·6 in. and pitch 0·3 in. This

corresponds to a total cooling area of only 3·0 sq. ft. per sq. ft. of internal surface, as against 4·6 and 5·0 on the barrels of the modern aero-engine cylinders of table 32. The fins on the specimen would, therefore, have been inadequate for cooling an engine cylinder, and it seems probable that had they been of the necessary greater depth and closer pitch to correspond with engine practice, the increase of heat dissipation with an oblique air-stream would have been less marked. The test specimen was of steel and carried only circumferential fins. The ends were protected by heat-insulated extensions provided with similar fins, so that the air-flow over the central heated portion was similar to that past a much longer specimen of similar form.

With the air-stream normal to the axis of the specimen the mean emissivity at 63·4 m.p.h. was 0·37 C.H.U. per sq. ft. per min. per deg. C., a higher figure than Gibson's best of 0·32 for a complete cylinder at the same air speed.

In the light of these figures we will now attempt some analysis of the separate parts played by the head and by the barrel of an engine cylinder in getting rid of the waste heat. It has been estimated in art. 61 (i), p. 232, that in an engine of compression ratio 5 : 1 the ratio of the heat to be got rid of through the walls of the head and of the cylinder barrel is about 1·2 to 1 if we regard the barrel as extending to the top of the piston travel and include in the head losses the heat given up to the exhaust-valve housing. The most common practice, nowadays, in air-cooled cylinders, is to have an aluminium alloy head screwed on to a steel barrel, as illustrated in fig. 4, p. 9. The top of the steel comes immediately above the limit of the piston travel, and the aluminium head is carried down outside the steel only so far as is necessary to make a screwed joint, about 1¼ in., on a cylinder of 5–6 in. bore.

The cooling areas on the aluminium head and the steel barrel for the two typical cylinders of table 32 are given in table 33, together with the estimated heat loss per sq. ft. from each part. It is to be noted that the average rate of heat dissipation per unit area is nearly as great on the barrel as on the head; but this, of course, does not mean that the average barrel temperature may be as high or higher than that of the head: quite the reverse. In the first place, there is ample evidence that the mean emissivity is normally higher on the barrel than the head, because of the very irregular form of the latter, and the inevitable pockets of dead air which contribute nothing to the cooling.

For the same reason the mean emissivity of cylinder heads may be expected to vary widely from one design to another, and it is unsafe to give figures for general application. On the question of the

TABLE 33

	Cooling area sq. ft.		*Heat dissipated C.H.U. per min.*		*Heat dissipated per sq. ft. per min.*		*Mean temperature difference for V_m = 100 m.p.h.*	
	Head	*Barrel*	*Head*	*Barrel*	*Head* (h_1)	*Barrel* (h_2)	*Head* ($h_1 \div$ 0·41)	*Barrel* ($h_2 \div$ 0·47)
							° C.	° C.
Jupiter .	5·24	3·47	465*	285*	89	82	217	174
Rapier .	1·80	1·35	176*	120*	98	89	239	189

* This subdivision between head and barrel allows for the fact that in each case a certain length of the barrel proper is included within the aluminium head. The total heat dissipated is in each case taken as 0·6 of the B.H.P., the same as in table 32.

cylinder barrel, on the other hand, it is possible to arrive at some fairly definite conclusions.

The results of Stanton's experiments upon copper fins round a dummy cylinder have already been given in equation (29) on p. 155 but may usefully be repeated here.

They are represented by

$$H = 0{\cdot}0057(1 + 0{\cdot}0075T_m)V_m^{0{\cdot}73},$$

in which H is the heat dissipated per sq. ft. per min. per deg. C. temperature difference, when the velocity of the free air-stream is V_m m.p.h. and T_m is the arithmetic mean of the absolute temperatures of the fin and air-stream.

In an air-stream of 60 m.p.h. and with mean fin and air temperatures of 157° C. and 27° C. the formula gives

$$H = 0{\cdot}425 \text{ C.H.U./sq. ft./min./deg. C.}$$

It happens that the steel test specimen with turned fins used in the American experiments, to which reference has been made above, was of almost exactly the same dimensions, both as regards the body, fins, and fin pitch, as in Stanton's experiments, and a comparison of the above figure with the emissivity found from the American report is of much interest. Under the same conditions of temperature and air speed the figure found from the American experiments is $H = 0{\cdot}355$.

In comparing these two figures it must be remembered that in Stanton's experiments the fins were uniformly heated by an electric current and must, therefore, have been working at nearly 100 per cent. efficiency.

The calculated efficiency of the steel fins on the American specimen, which were 1·52 cm. high and 0·15 cm. thick, comes out as

83 per cent. for $V_m = 60$ m.p.h., and allowing for this efficiency of the fins the emissivity from the Stanton formula would be 0·353. The closeness of the agreement with the observed figure must be regarded as fortuitous, seeing that in the American experiments the cylindrical base surface was playing some part in the heat dissipation, but the close agreement does give one confidence that the Stanton formula, with a factor included to cover the fin efficiency, may be applied equally to an actual cylinder barrel plus fins, although derived from experiments on circular fins by themselves. We are all the safer in this conclusion by reason of the fact that on actual cylinders the area of the fins is greater in relation to the base area than it was in the test specimen, where the ratio was 3. In modern engines the figure is about 5.

We may, therefore, conclude that for cylinder barrels the rate of heat dissipation in a free air-stream of velocity V_m m.p.h. parallel to the fin plane, will be given by the expression

$$H = 0{\cdot}0057(1+0{\cdot}0075T_m)\eta_f V_m^{0{\cdot}73}, \tag{32}$$

in which the fin efficiency η_f can be calculated by the method already given, provided the spacing is not so close as to allow interference of the boundary layers between the fins.

Turning now to the question of the cylinder head, we can make a rough analysis by employing the figure obtained by Gibson for the mean emissivity of a complete cylinder. As compared with that figure of 0·30 at $V_m = 60$ m.p.h. and a value of $T_m = 320°$ abs. we have the figure 0·355 for the barrel only from the American experiments at $T_m = 365°$ abs. To compare with 0·30 for the complete cylinder we, therefore, have 0·325 for the barrel only at the lower temperature. If we assume 30 per cent. more cooling area on the head than on the barrel, then the overall emissivity of 0·30 would be given by a combination of 0·28 for the head and 0·325 for the barrel.

Having increased these to correspond with an air speed of 100 m.p.h. we arrive at the figures in table 33 for the mean emissivity and the mean temperature difference above the cooling air, for the head and the barrel considered separately.

It is not suggested that the figures are truly representative for the two cylinders. The barrel temperatures are certainly too high. The figures, however, although admittedly only rough approximations, are sufficiently reasonable to justify the methods of analysis. One may conclude that the figure assumed for the total heat dissipated, 0·6 of the B.H.P., is about correct for the Jupiter engine, but that the allocation of this should have given a rather larger fraction to the head and less to the barrel. The figures for the Rapier engine suggest

that the total heat dissipated by the fins is less than 0·6 of the B.H.P. —as would be expected in view of its very high speed and high compression ratio. If it be taken as 0·55 of the B.H.P., the calculated temperature would be reduced to the level of those given for the Jupiter.

Although the mean emissivity over the barrel is not likely to vary much between one engine and another, yet that over the head fins, as already pointed out, is likely to show wide differences between one design and another; and the figures given must only be taken as roughly indicative of what is to be expected.

ART. 44. *The effect of cowling, gearing, and altitude conditions.*

Besides the differences due to fin design on the cylinder heads, there is also the effect of cowling and baffles to be considered. This is important, too, for the barrel fins, for in certain types of engine an effective air-flow can be maintained over a much larger portion of the barrel fins, with the help of suitably arranged baffles, than if the cylinder is simply exposed in the free air-stream. It has been found that the best cooling of a cylinder barrel was obtained when the baffles for controlling the air-flow were shaped to follow round the outside of the fins, leaving a clear space of about ¼ in. between the inner surface of the baffle and the outer edges of the fins. The baffle must only be carried round the cylinder barrel so as to subtend an angle of about 30 deg. beyond the plane at right angles to the direction of the air-flow. If the cowl is carried too far round on the down-stream side the quantity of air which can be got to flow within it rapidly falls off. The particular limits and curvatures which give the most effective air-flow on any particular engine are best settled by practical trials of the complete installation of engine and airscrew. In this connexion a recent comprehensive paper by Beisel and others[72] should be referred to.

If baffles are employed in connexion with the cylinder barrels they are always designed to improve the cooling conditions, and that is their justification. With the cowling of cylinder heads it is quite otherwise. Very often the conditions for cylinder heads are made more severe by the presence of the cowling, of which the justification is that it reduces the head resistance of the aeroplane as a whole. The conditions as regards both the velocity and direction of the air-flow over the cylinder heads, more especially of a radial engine, are profoundly affected by the form of cowling employed, and so, also, the necessary cooling surface per B.H.P. to avoid overheating.

Although the figures for the cooling area per B.H.P. in the two modern engines of table 32 came out very much the same, at 0·165

and 0·151 sq. ft., it is certainly unsafe to regard these or any other figures as generally applicable, except among engines of similar type and speed range, and with more or less similar cowling arrangements.

Besides the introduction of ring cowlings for radial engines, the advent of the geared engine has had a marked influence upon air-cooled cylinder design. For example, the most recent engines of types generally similar to the Jupiter show an increase of cooling surface up to 0·19 sq. ft. per B.H.P.; not because the cylinder temperatures were excessive before, but because for the same power output those cylinder temperatures must be maintained with a lower air speed over the fin surfaces. The large geared airscrew of low r.p.m. provides less slip-stream velocity over the cylinder barrels, and the heads suffer from the restricted air passages within the stream-lined cowl or Townend ring. Furthermore, when an engine is super-charged so as to maintain ground-level pressure in the induction pipe up to, say, 10,000 ft., the cooling problem at that height is much more severe than at ground-level.

If σ be the relative density of the air at 10,000 ft. and if V and V_0 are the speeds in level flight at full engine power, there and at ground-level, then V, V_0 and σ are closely related by the equation

$$V/V_0 = 1/\sigma^{\frac{1}{2}}. \tag{33}$$

If it were possible for the engine to maintain constant revolutions as well as a constant torque below the supercharged height, then the power output would be constant and the ratio of the top speeds in level flight would be $1/\sigma^{\frac{1}{3}}$. An airscrew of fixed pitch, however, will prevent the maximum permissible r.p.m. being reached below the supercharged height, and the ratio V/V_0 is, therefore, greater than it would be if constant power were maintained, and is very nearly $1/\sigma^{\frac{1}{2}}$ as given in equation (33).

Since $V_0 = V\sigma^{\frac{1}{2}}$ it follows that the A.S.I. reading, airscrew efficiency, and V/n are all unchanged by the change of height, and therefore the ratio of the amounts of heat to be dissipated, which are proportional to the power, are given by

$$\frac{H}{H_0} = \frac{P}{P_0} = \frac{n}{n_0} = \frac{1}{\sigma^{\frac{1}{2}}}. \tag{34}$$

The ratio of the rate of heat dissipation per unit difference of temperature, at 10,000 ft., to that at ground-level, will be proportional to the air density, and is, therefore,

$$\sigma\left(\frac{V}{V_0}\right)^{0\cdot73} = \frac{\sigma}{\sigma^{0\cdot365}} = \sigma^{0\cdot635}.$$

The temperature of the air in the standard atmosphere is $-5°$ C. at 10,000 ft. as compared with $+15°$ C. at ground-level, so that if we take 180° C. as the mean temperature of the cooling surfaces we may say that there will be a gain in the rate of cooling of 12 per cent. on account of the lower temperature.

At 10,000 ft. $\sigma = 0{\cdot}738$, and, therefore, the ratio of the rate of cooling there, to the rate at ground-level, will be

$$1{\cdot}12 \times 0{\cdot}738^{0{\cdot}635} = 0{\cdot}92,$$

while the amounts of heat to be dissipated are in the ratio

$$\frac{1}{\sigma^{\frac{1}{2}}} = \frac{1}{0{\cdot}86} = 1{\cdot}16.$$

It is clear, therefore, that an engine which was just adequately cooled when giving full power in level flight at ground-level, would be badly undercooled at 10,000 ft.; for not only would the heat to be got rid of be 16 per cent. greater, but the rate of heat dissipation for the same cylinder temperature would be 8 per cent. less.

It may be objected that most engines are amply cooled at full speed in level flight. The same general comparison, however, can be applied to climbing conditions, for the best rate of climb will correspond as a rule to a nearly constant A.S.I. reading at all heights: so that, once again, the ratio of true speeds and of engine r.p.m. will be in the ratio $1/\sigma^{\frac{1}{2}}$. But all speeds being much lower, the cooling problem is proportionately more severe than in level flight.

The desirability of maintaining the same cylinder temperatures between top speed and climbing conditions has called for some degree of variability in the engine installation: in other words, for designs of variable cowling. With completely enclosed cylinders, the quantity of air reaching them can be controlled either at entry to, or exit from, the cowling, thus preventing over-cooling at top speed if the fin-area is adequate for climbing conditions. Not that prevention of over-cooling at top speed is in itself the important thing, but rather the possibility of improving the streamline form of the cowling and fuselage or nacelle, and of reducing the amount of general turbulence, when the size of the air passages needed for cooling are reduced.

ART. 45. *Summary and conclusions.*

As a result of supercharging, gearing, and the introduction of cowling, the air-cooled engine designer has in recent years been forced to crowd more and more fin area on to his cylinders. As compared with the earlier engines, a greatly increased power output

per unit of swept volume has been achieved, either by higher speeds, higher B.M.E.P.s, or by both together. The first result of this was a large reduction of the cooling surface per B.H.P., but the reduced area was adequate, because of the higher forward speeds attained and better cylinder-head design. Then came the geared engine, the Townend ring, and the supercharged engine, each calling for more and more cooling surface even at the same maximum power output, because of the less favourable conditions which the engine designer has been called upon to meet.

These increases of cooling surface could only be provided by designing deeper fins, more closely spaced, and the result has been that, in spite of the ever-increasing power output per unit of cylinder capacity, the cooling area provided per B.H.P., after falling from about 0·25–0·30 to 0·17 sq. ft. per B.H.P., now shows signs of increasing again to nearly 0·20.

From what has been set out during the course of this study of air-cooling it will be obvious that a mere increase of the fin area per unit of swept volume is not, by itself, sufficient to improve the cooling of an engine. What is essential is to achieve an increase of *useful* fin area, or, in other words, to increase the fin area without at the same time reducing the average fin efficiency at a given air speed. To do this successfully, without an endless series of hit-and-miss experiments, a knowledge of first principles is essential, for it is only by having a clear mental picture of the true nature of the flow of air near the cooling surfaces that pitfalls will be avoided.

Calling in aid the results of theory and experiment, the following conclusions may perhaps be set down as landmarks; but each one may require qualifying or expanding after the necessary detailed study of any particular problem.

(1) A rate of dissipation of heat from the outside of the cylinder walls equivalent to 0·5–0·6 of the B.H.P. must be achieved, by providing a sufficient area of metal surface which is effectively swept by the cooling air-stream.

(2) Close to every part of the cooling surfaces swept by a tangential air-stream there is a 'boundary layer', of a thickness varying from a few thousandths to a tenth of an inch, and within this layer the air velocity falls until it is zero at the surface itself. Over a certain portion of the surface, near the leading edge of a fin, the flow within the boundary layer is laminar, but changes later to the turbulent type. There is a sudden thickening of the boundary layer at the point where the flow within it becomes turbulent. So long as the flow within the boundary layer is laminar a tangential air-stream is less effective for cooling a fin, but, on the other hand, on the up-

stream side of a cylinder the laminar flow over the fins is counterbalanced by the direct impingement of the air upon the barrel.

(3) There is a fundamental relation between skin friction and heat transmission which fixes a minimum drag that must be associated with getting rid of a certain amount of heat. This friction associated with heat transmission must constitute additional drag on an aeroplane unless the heat can be dissipated from some surface, such as a wing surface, which must necessarily be there in any case. With an air-cooled engine there must always be extra drag associated with the cooling process.

(4) Cooling fins may become inefficient if they are too high or too thin, on account of their tips being over-cooled compared with their roots. The most efficient fin would have hollow curved sides; but a triangular tapered, or even a rectangular, fin loses little in efficiency compared with the ideal shape. The most convenient form to manufacture is a slightly tapered fin; and any fin which is stiff enough to withstand manufacture and rough handling is unlikely to have an efficiency less than 60 per cent. provided there is an adequate airflow over it. In consequence it will always pay to have fins as thin as their manufacture will allow, and spaced as close as will allow of a proper air-flow between them.

(5) If the pitch of the fins is small enough to cause the boundary layers of two neighbouring fins to interfere with one another over any considerable area, a rapid fall in the rate of heat dissipation will occur. The minimum allowable clear space between the fins appears to be of the order of 0·15 in. when the air has to flow between the fins for a distance of 6 in. from the leading edge in the down-stream direction.

(6) It is possible to estimate with fair accuracy the rate of heat dissipation from a finned cylindrical barrel subject to a free airstream at right angles to its axis; but the rate can be improved by judicious cowling to increase the effective area of the fins.

(7) It is possible, and even more important, to improve the airflow over a cylinder head by carefully designed cowling. Apart from this, it is necessary to cool certain vital parts, such as those near the exhaust valve, by a directly impingeing air-stream. On the other hand, this use of the air available must not be allowed to lower too much the average tangential velocity over the rest of the cooling surfaces.

VIII

LIQUID COOLING. RADIATORS

ART. 46. *Heat transfer and the flow of air in pipes.*

It was stated in art. 40 that of the 20–25 per cent. of the total heat supply of an engine which is dissipated to the surrounding air, otherwise than in the exhaust gases, not more than 15–20 per cent. would be conducted through the walls of the cylinders. The rest would be conducted away to the crankcase, or removed from the undersides of the pistons by lubricating oil, and would ultimately be dissipated from an oil cooler and from the exterior of the crankcase. In a liquid-cooled engine the proportion of the total heat which is lost to the cylinder walls and pistons will be substantially the same as in an air-cooled, and will be equivalent, in an efficient, high-speed engine, to between 50 and 60 per cent. of the B.H.P., depending on the speed and on the design of the cylinder heads and exhaust-valve ports. The whole of this, however, does not have to be dealt with in the main radiator, for there may be considerable loss of heat direct from the piping and outer walls of the cylinder jackets before the liquid reaches the radiator. Tests on a 12-cylinder engine designed about 1916, at 2,000 r.p.m., showed that when the engine was run with a normal air-fuel ratio and closely covered in to prevent any direct loss of heat from the cylinders, an amount equivalent to 60 per cent. of the B.H.P. passed to the radiator in the cooling water; and that this dropped to 52 per cent. when the engine was uncovered and run on a rich mixture, with a wind of 60 m.p.h. blowing over it. At other wind speeds the heat passing to the radiator was in proportion.

In modern engines of higher speeds the amount of heat passing to the jackets is less, and figures as low as 42 per cent. of the B.H.P. have been observed on normal aero-engines. It is of interest that Railton[40] gives the amount of heat passing to the radiator of the record-breaking 'Blue Bird' car of 1933, fitted with one of the Rolls-Royce engines designed for the Schneider Trophy of 1931, as 38 per cent. of the B.H.P. at the maximum speed of 272 m.p.h. and 3,000 r.p.m. of the crankshaft.

The factors to be considered in arriving at the optimum design for an aircraft radiator are too many to be included in any general theory. The drag, the rate of heat dissipation, the weight of the radiator and of the contained water, can all be made the subject of experimental investigation in different designs, and certain general conclusions can be embodied in a calculated 'figure of merit' for

radiators of the same general type; but the figure of merit itself will depend upon the type of aeroplane in which the radiator is to be used, and, even then, a figure based upon the above-stated essentials of radiator performance may lead to a design which is undesirably bulky or difficult to install.

Accurate experimental data are available[41] upon the characteristics of a number of types of radiator. It is proposed here only to state in general terms the principles which underlie the transfer of heat from water to air in a radiator, so that when consulting the original sources for fuller information the reader may see the detailed results of experiment against a proper background.

It was shown in the last chapter that, according to the simple theory of Osborne Reynolds in which the boundary layer is not taken into account, the rate of heat transmission H and the force of skin friction R, are related by the equation

$$H = \frac{RK_p}{V_m}(T_s - T_m),$$

in which T_s and T_m are the mean temperatures of the hot surface and air respectively, and V_m is the mean velocity of air-flow. For the particular case of the flow of air through pipes the agreement or otherwise of theory and experiment has been very fully examined, notably by Stanton and Pannell.[42] In the latter's experiments, air was drawn through a heated metal pipe 4·88 cm. diam. and measurements were made of the temperature rise, fall of pressure, temperature distribution, and velocity distribution, of the air over a certain length of the pipe. The results are summarized in table 34,

TABLE 34

Results of Pannell's experiments on the rate of heat transfer to air flowing in a pipe.

$V_{max.}$ cm./sec.	V_m cm./sec.	T_s °C.	T_m °C.	R dynes/sq. cm.	H cal./sq. cm. (observed)	$(T_s - T_m)\frac{RK_p}{V_m} = H_c$ cal./sq. cm.	H/H_c
688	542	35·3	24·8	1·32	$0{\cdot}905 \times 10^{-2}$	$0{\cdot}612 \times 10^{-2}$	1·48
1,483	1,200	37·4	23·7	5·13	$2{\cdot}08 \times 10^{-2}$	$1{\cdot}406 \times 10^{-2}$	1·48
1,820	1,482	38·0	23·7	7·67	$2{\cdot}595 \times 10^{-2}$	$1{\cdot}775 \times 10^{-2}$	1·46
2,760	2,180	43·0	27·3	14·97	$3{\cdot}66 \times 10^{-2}$	$2{\cdot}585 \times 10^{-2}$	1.42

from which it will be seen that, while the measured rate of heat transfer, H, was very closely proportional to

$$(T_s - T_m)\frac{RK_p}{V_m}$$

over a wide range of V_m, it was between 40 and 50 per cent. greater than the value given by theory.

As would be expected from what has been said of boundary-layer conditions in Chapter VI, both the rate of heat transfer and the surface friction are affected for some distance along a pipe by the entry conditions. Stanton's experiments have shown that after

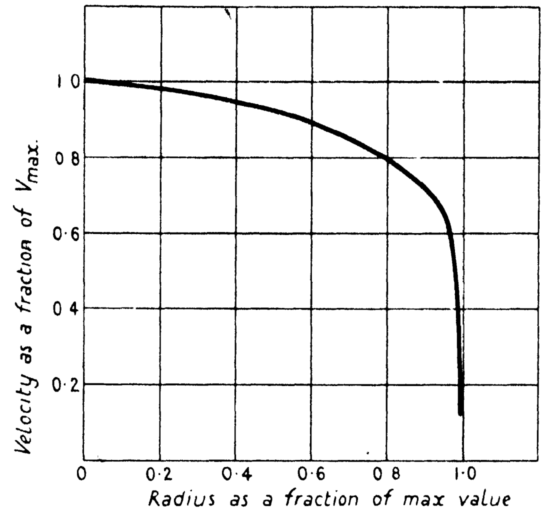

FIG. 64. Variation of velocity with the distance from the centre of a circular pipe, after the flow has become steady, at distances of more than 100 diameters from the entry.

traversing a length of pipe equal to about 100 diameters from the entry these effects are eliminated, and that the ratio of the mean to the maximum flow velocity across a cross-section of the pipe has by then settled down to a constant value. In Pannell's experiments the measurements were made over a length of 61 cm. beginning 550 cm. from the entry of a pipe less than 5 cm. diam., so that they represent conditions in which the boundary layer was fully turbulent. Under these conditions the distribution of velocity across a diameter of the pipe is as shown in fig. 64.

From the figures of table 34 it will be seen that under these conditions the rate of heat transfer is very closely proportional to V_m and the friction, therefore, to V_m^2. Near the entry of the pipe, on the other hand, where the boundary layer would be laminar, we have already seen in Chapter VI that the rate of heat transfer and the surface friction will be closely proportional to $V_m^{0.5}$ and $V_m^{1.5}$ respectively. In a radiator tube, where laminar merge into turbulent conditions after a certain distance, it has been found that the rate of heat transfer in-

creases rather less rapidly than the air velocity. In fig. 65 are shown the measured rates of heat dissipation in horse-power at wind speeds from 40 to 90 m.p.h. for two series of radiators, the first series having tubes of round, and the second of hexagonal, cross-section. In each case the tubes are 'bulged' at their ends to a hexagonal shape for soldering up. The curves are all slightly convex upwards, showing a rate of heat dissipation less than proportional to the wind

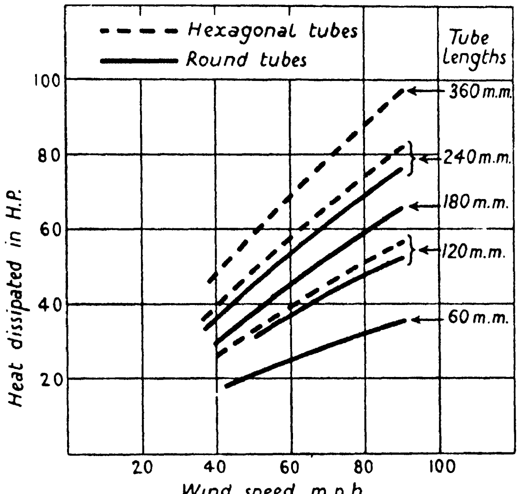

Fig. 65. Rates of heat dissipation per sq. ft. of frontal area at different air speeds, for radiators with round tubes of 10 mm. bore and of different lengths. Mean temperature difference 66·6° C. (120° F.).

speed. The curves represent radiators having tubes of the same cross-section but of different lengths. It will be seen that equal increments of length produce diminishing increments of heat dissipation, so that the mean rates of dissipation per unit area of cooling surface are as given in table 35, the comparative figures being taken

TABLE 35

Rate of heat dissipation in h.p. per sq. ft. of cooling surface with tubes of different lengths.

Tube length mm.	*Round tubes 10 mm. diam.*	*Hexagonal tubes 10 mm. across flats*
60	1·37	..
120	1·03	0·97
180	0·89	..
240	0·79	0·71
360	0·60	0·57

for a wind speed of 60 m.p.h. and a mean temperature difference between the air and water of 120° F. It must of course be remembered that the wind speed of 60 m.p.h. does not, as in Pannell's experiments, represent the mean speed through the tubes. If the mean air speed through the tubes had been constant, then, apart from the effect of the laminar flow at entry, each increment of tube length would add to the heat dissipation in proportion to its area, for the same mean temperature difference. In the experiments upon complete radiators there is a certain amount of heat dissipated from the outside of the tube block and from the 'dead area' of the front surface, i.e. the total frontal area less the sum of the inside areas of cross-section of the tubes. This latter will be more or less constant for the different lengths of tube, but apart from this, when the radiator is simply set up in an air-stream of 60 m.p.h. the mean air speed through the tubes will fall off as their length increases, and it is this effect which is illustrated by the figures of table 35.

According to the data given in art. 36 the boundary layer near a flat plate would become turbulent, for $V_m = 60$ m.p.h., at a distance of about 14 cm. from the leading edge. After that point the thickness of the layer would increase from about 5 mm. One cannot safely make quantitative deductions for a pipe from the known conditions near a flat surface, but one may safely argue that for a pipe 10 mm. diam. and more than 150 mm. long the maximum velocity through it will never reach that of the external free air-stream; and the result of any increase of length will be to reduce the maximum velocity in the pipe to something less than the velocity in the boundary layer at a distance from a surface equal to the radius of the pipe.

The thickness of the boundary layer in the turbulent region increases in a nearly linear manner with distance in the down-stream direction (see fig. 55), and a way of maintaining the ratio of the internal to the external air velocity in a long pipe would clearly be to increase the pipe diameter in proportion to its length. In a series of radiators of a given frontal area, built up of tubes of different diameters but of a constant L/D ratio, the total cooling surface would be roughly constant, and experiment shows that the heat dissipation per sq. ft. of surface is also constant. In other words, as the tube length is increased, each increment of length is equally effective because the air speed through the tube is maintained; in contrast to the state of things when a tube of constant diameter is lengthened. The figures in table 36 illustrate this point. They are the results of experiments upon three radiators, each of about 1 sq. ft. frontal area, when tested in a free air-stream of 60 m.p.h. at a mean temperature difference of 67° C. Although there is little to

TABLE 36

Rates of heat dissipation from three radiators with tubes of a constant length/diameter ratio, in h.p. per sq. ft. of cooling surface when tested in a free air-stream of 60 *m.p.h. and at a mean temperature difference of* 67° *C.*

	Radiator No.		
	(1)	(2)	(3)
Frontal area	0·996 sq. ft.	0·962 sq. ft.	0·960 sq. ft.
Tube size, length and diameter	85 × 7 mm.	120 × 10 mm.	180 × 15 mm.
Cooling surface	33·0 sq. ft.	35·1 sq. ft.	35·9 sq. ft.
Heat dissipation h.p. per sq. ft. of cooling surface	1·00	1·03	1·00
Heat dissipation h.p. per sq. ft. of frontal area	33·1	37·5	37·4

choose between them in regard to heat dissipation per sq. ft. of cooling surface, the long tube type is somewhat superior per unit of frontal area, because the cooling surface per sq. ft. of frontal area is 37·4 sq. ft. in radiator no. 3, as compared with 33·1 sq. ft. in no. 1.

The superiority of the long tube radiator is much more marked in regard to the question of drag, to be considered in the next article. In this respect radiator (1) of table 36 is some 12 per cent. worse than no. 3, by reason of the greater proportion of the 'dead' part in the total frontal area when small tubes are used. On the other hand, the long tubes of large diameter, although requiring rather less frontal area for a given rate of heat dissipation, make a bulky radiator, very difficult to instal unless slung, somewhat awkwardly, beneath the fuselage. It will readily be understood, moreover, that where round tubes are used, a larger diameter leaves more space to be filled by water. In radiator no. 3 of table 36 the weight of water carried within the radiator was about 6 per cent. greater than in no. 1.

ART. 47. *The drag of radiators.*

It was pointed out in art. 40 that if the waste heat from an engine could be dissipated from surfaces which themselves formed essential parts of an aeroplane, then no engine power would be absorbed in effecting the necessary heat dissipation. A wing-surface radiator approaches such an ideal, although there will almost always be pipes and other special features connected with the cooling system, of which the extra drag must be charged to its account. Even in the Schneider Trophy seaplanes, where all possible obstructions were eliminated and the outer form of the wing surface, which was also the radiator, was unaltered, it was found necessary to circulate air inside the wing, by means of small scoops, and this produced some extra drag.

It was further shown in art. 40 that, failing the use of essential surfaces of the aeroplane for dissipating heat, it is possible to conceive the special radiating surfaces as being so perfectly streamlined that nothing but drag due to skin friction would be involved. And it was calculated that in this case a rough figure for the extra drag due to the cooling surfaces was 1·5 per cent. of that of the whole aeroplane. As compared with this figure, it was estimated that a tubular radiator on a well streamlined aeroplane was responsible, in one particular experiment, for about 9 per cent. of the total drag. The reason for this wide disparity is, of course, that besides the skin friction on the effective cooling surfaces (i.e. the internal surfaces of the tubes) there is the direct impact on the 'dead' part of the frontal area, as well as on the supporting frame and other accessories of the actual tube-block. Apart from the direct contribution of these to the total drag, there is also the general turbulence set up by deflexion of the air-flow in the neighbourhood of the radiator, and this increases the drag of other parts of the aeroplane.

The drag of the various radiators discussed in the last article was measured at 40 and 60 m.p.h., both for the tube-block only and for the radiator complete with its frame and the essential pipes. In what follows, drag is discussed in terms of that for 1 sq. ft. of the tube-block at 60 m.p.h. The frame and accessories added between 40 and 50 per cent. to the drag of the tube-block by itself. The drag at other speeds was found to be accurately proportional to V^2.

TABLE 37

Rates of heat dissipation, and drag, each per sq. ft. of frontal area for two series of radiators, (a) *with tubes of a constant diam.* 10 *mm. and* (b) *with tubes of a constant ratio of length : diameter* = 12.

Tubes of constant diameter D = 10 mm.					*Tubes of constant L/D = 12*			
Tube length mm.	*Cooling area sq. ft.*	*Heat dissipation per sq. ft. Frontal*	*Drag per sq. ft. Frontal lb.*	*Drag per h.p. dissipated lb.*	*Cooling area sq. ft.*	*Heat dissipation per sq. ft. Frontal*	*Drag per sq. ft. Frontal lb.*	*Drag per h.p. dissipated lb.*
60	18·0	23·6	3·21	0·136				
85	..	..	..	..	33·0	31·0	4·64	0·15
120	35·1	35·3	4·13	0·117	35·1	35·3	4·13	0·117
180	50·6	43·9	4·93	0·112	35·9	34·5	3·96	0·115
200	55·9	50·1	5·30	0·106				
240	67·7	52·2	5·52	0·106	*Hexagonal tubes* 10 *mm. across flats*			
300	90·9	55·9	5·95	0·106				
360	109·6	61·9	6·40	0·104	120·4	64·4	6·18	0·096
420	127·8	64·9	6·78	0·104				

In table 37 the rate of heat dissipation and the drag, each per sq. ft. of frontal area, are shown for a wide range of tube-lengths, firstly

for a series with tubes of constant diam. $D = 10$ mm. and secondly for three blocks of tubes with a constant L/D ratio of 12. In order to make the figures for drag and for heat dissipation properly comparable, the rates of heat dissipation previously given for complete radiators have been corrected so as to correspond with 1 sq. ft. of the tube-block only. For the third radiator in the table, with 120 mm.

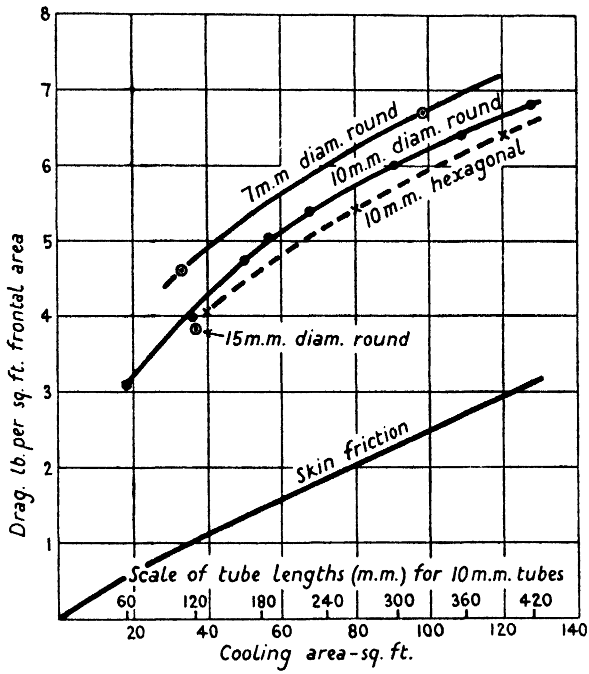

FIG. 66. The variation of the drag of different types of tube-block with the cooling area, measured per sq. ft. of frontal area in a wind of 60 m.p.h.

tubes (no. 2 of table 36), those parts of the radiator other than the tube-block accounted for 2·4 h.p. or about 6 per cent. of the total heat dissipation at 60 m.p.h. For other lengths of tube the correction was estimated to vary from 1·8 h.p. at 60 mm. to 3·0 at 200 mm., and 4·5 at 420 mm.

In fig. 66 the drag of different types of tube-block at 60 m.p.h. has been plotted against the area of the cooling surface, for tubes of different diameters,* together with the corresponding drag if this were due to skin friction only. The curves of measured drag and of calculated skin friction become very closely parallel for the longer

* The curves include the results of a few experiments not quoted in table 37.

tube-blocks, the vertical distance between the curves then representing the nearly constant drag due to the dead part of the frontal area.

In table 37 the drag in 'lb. per h.p. dissipated at 60 m.p.h.' is also given as being a figure which, from the limited point of view of drag and heat dissipation only, may be regarded as a rough figure of merit for the various types. Other no less important factors are the weights of the different types of tube-blocks and of the water contained by them; the effect of shutters, which can be made to reduce the drag substantially while reducing the heat dissipation to that appropriate for cruising conditions; and the influence of yaw, i.e. of the airstream not being parallel to the radiator tubes. As regards the last point it may be said at once that some increase of heat dissipation can be obtained for small angles of yaw, reaching a maximum of about 10 per cent. at 30–35 degrees, but that the rapid increase of drag would never allow this method of increasing the heat dissipation to be used with advantage. Questions of weights and of the effect of shutters it is not proposed to discuss in any detail. They can only be treated adequately in conjunction with the details of particular designs, which are outside the scope of this book. For all such matters, therefore, the reader should consult the original sources[41] cited.

To illustrate the effect of tube-length and diameter one may usefully take as a starting-point in table 37 the block with tubes 120 mm. long and 10 mm. diam. This occurs in both halves of the table, and is the same as that in radiator no. 2 of table 36. It will be seen that the drag per h.p. dissipated is very substantially lower than for either of the two shorter blocks, the one with 10 mm. and the other with 7 mm. tubes. As between the large and small diameter tubes, the latter provide a larger 'dead' fraction of the frontal area, and therefore a greater drag, in spite of their shorter length and rather less cooling surface. For the series of 10 mm. tubes there is at first a considerable reduction of the drag per h.p. as the tube-length is increased. Beyond a length of 240 mm. there appears to be little gain, and moreover there is a practical disadvantage in using very long tubes, on account of the difficulty of keeping them perfectly straight, and the water passages uniform, during assembly. In their favour, however, it must be noted that where shutters are used it is of vital importance to keep down the frontal area of the radiator, upon which the drag will largely depend when the shutters are more than half closed. The increased heat dissipation per unit of frontal area shown in the table for the long tubes is therefore specially desirable in a fast aeroplane with high-power engine; for the radiator which gives adequate cooling for such an engine under climbing conditions

will inevitably overcool it in level flight unless the effect of the increased air speed is corrected by shutters.

The data in table 37 were all for radiators with round tubes, although expanded to a hexagonal cross-section at their ends for soldering up, the hexagon being of such a size as to leave 1 mm. clearance for the water-flow between the tubes. The hexagonal ends of a 10 mm. tube would therefore be 11 mm. across the flats.

The lowest of the three curves of measured drag in fig. 66 refers to a series of trials of tube-blocks composed of tubes of hexagonal section throughout, 10 mm. across the flats and with ends identical with the round 10 mm. tubes. The cross-section available for water-flow in such a radiator will clearly be uniform and 1 mm. wide, and there will be a substantial saving in the weight of the contained water; thus for a tube-block of 1 sq. ft. area composed of tubes 400 mm. long the contained water for 10 mm. hexagonal, as compared with 10 mm. round, tubes would be 13·5 lb. as against 20 lb. The drag of the hexagonal tubes was uniformly lower by 0·2 lb. per sq. ft. of frontal area than that of the round ones, and since the dead area was the same, the lower drag was no doubt due to the maintenance of a slightly larger cross-sectional area throughout the length of the tubes. Comparison of the two radiators with 360 mm. tubes in table 37 shows that per sq. ft. of frontal area the hexagonal tubes show a 3½ per cent. greater heat dissipation, but that this is due to the greater cooling surface available per unit of frontal area. The heat dissipated per sq. ft. of cooling surface was actually about 5 per cent. less in the hexagonal tubes.

Art. 48. *The derivation of a 'figure of merit'.*

In order to indicate the lines along which practical figures of merit for radiators may be developed it is proposed to summarize here the empirical formulae which have been derived from experiment, although the treatment will not aim at being complete enough for practical application.

(*a*) *Drag per unit of frontal area.* For the purpose of general analysis it has been found convenient to express the drag in lb. per sq. ft. of frontal area in the form CS^n, where S is the 'cooling area ratio' of the total cooling area to the frontal area. With the constants given in table 38 the error in this formula is less than ½ per cent. between $S = 60$ and $S = 130$, about 1 per cent. at $S = 50$, and 3 per cent. at $S = 40$.

(*b*) *Heat dissipation and air speed.* It has been shown in fig. 65 that the rate of heat dissipation is rather less than proportional to the

TABLE 38

Values of the constants C and n in the formula CS^n for the drag in lb. per sq. ft. frontal area.

Diameter and type of tube	*C*	*n*
10 mm. round	2·43	0·38
10 mm. hexagon	2·03	0·41

free air-stream velocity, and within the velocities tested (up to 90 m.p.h.) it may be taken as being proportional to the function

$$1-e^{\cdot 0\cdot 0026V}$$

for 10 mm. tubes. The advantage of using a law of this type, as compared with a simple function of V raised to some power x, is that it is consistent with a limiting rate of heat dissipation at very high speeds, and this condition must hold for a radiator with a finite rate of water-flow through it.

(*c*) *Heat dissipation and cooling area.* The following relation between the rate of heat dissipation H per sq. ft. of frontal area at 90 m.p.h., and the cooling area ratio S, has been found to hold over the whole range of the experimental results reviewed, with an error of less than 3 in 1,000:

$$H = A(1-e^{kS+b}).$$

The values of the constants A, k, and b are as follows:

Type of tube	*A*	*k*	*b*
Round	111	−0·0146	−0·059
Hexagonal. . .	116	−0·0143	−0·022

(*d*) *Total weight.* The total weight of the tube-block and contained water in lb. per sq. ft. of frontal area may be expressed by the function

$$\alpha+\beta l$$

in which l is the overall length of the tubes in mm. and α and β are constants which must be chosen appropriate to the type and diameter of the tubes.* For 10 mm. tubes their values are as follows:

Type of tube	α	β
Round . . .	0·258	0·122
Hexagonal . .	0·445	0·114

In order to obtain a comparison of the merits of different types of radiators from the point of view of their effect upon aeroplane per-

* See table 9, *R. and M.* 952.

formance, it is necessary to combine their characteristics of drag and weight for a given rate of heat dissipation. The relative importance of 1 lb. increase of drag or weight will depend upon the type of aeroplane, and upon whether the effect upon top speed or rate of climb is in question.

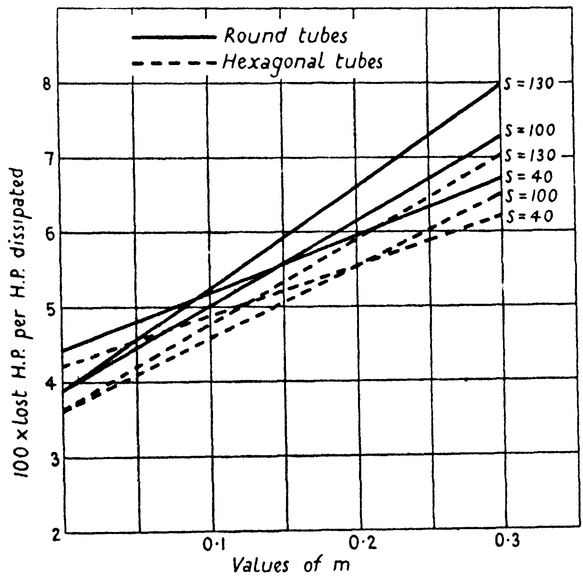

FIG. 67. Lost horse-power per h.p. dissipated per sq. ft. of frontal area for radiators with 10 mm. tubes of various lengths. The straight lines represent the function

$$\frac{(mw+r)V}{3{\cdot}75h}.$$

If we suppose that in certain circumstances an additional weight of 1 lb. will affect the performance, either in speed or climb, by the same amount as an additional drag of m lb., and that a radiator of total weight w lb. will give, at a speed V, a rate of heat dissipation h and drag r, then the 'effective drag' of the radiator per h.p. dissipated will be

$$\frac{mw+r}{h}\text{ lb. per h.p.}$$

and at the speed V the lost horse-power per h.p. dissipated is

$$\frac{mw+r}{h}\,\frac{V}{375},$$

V being in m.p.h.

As an example of the kind of values which m may assume, one may quote the case of an aeroplane with a top speed of 130 m.p.h.

at 15,000 ft., for which condition m would be 0·11, while for the same aeroplane climbing at 15,000 ft. m would be 0·31. For an aeroplane of a lower top speed the difference between the value of m when climbing and flying level would be less marked.

Since m must be chosen appropriate to each particular case it is convenient to plot the expression for the percentage loss of h.p. against m as the independent variable. It will clearly be a linear function of m. In fig. 67 the percentage loss of h.p. is shown plotted against m for 10 mm. tubes and for several values of the cooling surface ratio S, the values of h, r, and w having been derived from the empirical formulae given above and from the tables of art. 47. It should be noted that, in the formula represented in fig. 67, V is the true speed of the aeroplane, whereas the speed at which r and h have to be calculated is the wind speed at the radiator, which will probably be in the slip-stream from the air-screw.

On a diagram such as fig. 67 it is evident that the lower of two points for any given value of m represents the better radiator, and the superiority of the hexagon tube for a given value of S is clearly demonstrated. On the other hand, the round tubes at $S = 40$ are superior, when m is over 0·21, to the long hexagon tubes for which $S = 130$.

Art. 49. *The 'suitability factor' of a radiator for a given aeroplane.*

The provision of a radiator of the correct size for a given road vehicle is a fairly straightforward problem. The radiator must be large enough, and the fan speed sufficient, to prevent the water boiling on any but the steepest and longest hills. If the radiator is unnecessarily large for normal running, so that the temperature of the water seldom approaches the boiling-point, the results will not be serious, for the engine probably has an ample margin of power and the only effect of its being overcooled will be that the mechanical efficiency will be on the low side on account of a high oil viscosity, and the fuel economy expressed as miles per gallon will suffer slightly in consequence. There is, on the other hand, the compensation of smoother running usually met with in an overcooled engine.

The design of the radiator for a given engine in the air is a much more complex problem, both because the variety of conditions met with is greater and because the need to work under optimum conditions makes it essential to assess the effect of each variation.

Suppose, for example, that it is required to examine the suitability of a certain radiator experimentally. Under what conditions shall it be tested? When flying level at cruising speed? or at maximum level speed with fully open throttle? or when climbing at full

throttle? An obvious reply might be that it must be tested under the most arduous conditions it will have to meet, and these would be, broadly speaking, while climbing at the maximum rate at full throttle; for the engine would then be giving nearly its full power, while the forward air speed available for cooling would be only about 50 per cent. of the maximum level speed.

There is, however, the very serious difficulty about a test at the maximum rate of climb that the engine power, the maximum allowable water temperature, the forward speed, and the rate of heat dissipation for a given water temperature and forward speed, *all* vary with the height, and the ideal radiator for a given aeroplane and engine would be different at every height.

The ideal radiator would maintain the water always at the maximum allowable temperature, which is usually taken as being 3° C. below the boiling-point at any height. This would ensure that the engine was working always at its highest mechanical efficiency, and therefore at its minimum fuel consumption per B.H.P.; and furthermore it would correspond to the minimum size of radiator necessary for the engine, and therefore to minimum drag.

We must now see how a 'suitability' figure for the radiator can be arrived at which will express its cooling capacity in relation to what is required of it, and will at the same time be so far referred to standard conditions as to express the comparative merits of different radiators for any given purpose. What is required is a comparison of the actual rate of heat dissipation of the radiator, in given atmospheric conditions, with the maximum rate which it *could* provide if the maximum demand were to be made upon it; 'maximum demand' being defined by the engine being at full throttle and the air velocity through the radiator a minimum.[35]

Imagine the aeroplane flying at a height H in the standard atmosphere, where p and σ are the relative pressure and density, θ_a the air temperature, θ_m the maximum allowable water temperature, that is, 3° C. below the boiling-point at the pressure defined by p, θ_R the mean radiator temperature, and V the forward speed.

Assuming, for the present, that the engine is normally aspirated so that the B.H.P. is nearly proportional to p (see art. 74), then to a first approximation we may assume that the heat which passes to the radiator is equal to the same fraction of the B.H.P. at all heights, and therefore that the heat to be dissipated per sec. is also proportional to p.

The heat dissipated by the radiator per second will be proportional to $(\theta_R-\theta_a)\sigma V$, and the flying speed while climbing will be nearly proportional to $\sigma^{-\frac{1}{2}}$, this being the condition of constant 'A.S.I.

reading'. The rate of heat dissipation is therefore proportional to $(\theta_R-\theta_a)\sigma^n$, where n is equal to, or more often somewhat greater than, $\frac{1}{2}$. The maximum possible rate of heat dissipation would, correspondingly, be proportional to $(\theta_m-\theta_a)\sigma^n$.

The ratio $(\theta_m-\theta_a)/(\theta_R-\theta_a)$ of the maximum possible dissipation rate to its actual value at any height is accepted as the 'suitability' figure for the radiator, but in order to understand how it can be arrived at we must first examine how the heat capacity of the water system, and the

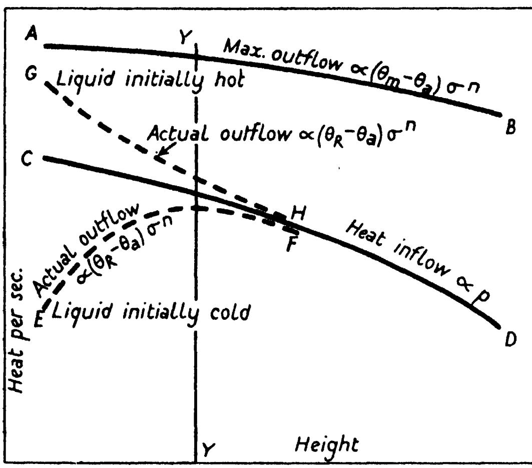

FIG. 68. The heat balance of a radiator in a climbing aeroplane at different heights.

starting conditions of the climb, will affect the observed temperatures during a test. As already pointed out, we are faced by the contradiction that our suitability figure is to refer to definite conditions as regards height, etc., and at the same time is to be based upon the conditions characteristic of a climbing aeroplane, wherein all the observable factors continuously vary. We must allow, too, for the fact that a test will not normally be made in a standard atmosphere.

In fig. 68 let the ordinates of the line AB represent the maximum possible rates of heat dissipation at any height, and those of CD the rate of heat-flow from the engine to the cooling water, proportional to p. Each of these quantities are quite definite for any height, and it can be shown that the first falls off less rapidly than the second as the height increases. Now as regards the actual heat dissipation we have to consider two alternatives: the water may be nearly cold as the aeroplane leaves the ground, in which case there will be a rapid rise of the rate of dissipation represented by the dotted line EF as the water heats up; or alternatively the water may have been allowed to get

very hot before leaving the ground, so that at first there is a fall in the temperature and in the rate of dissipation along the line *GH* while the maximum possible rate is at its greatest.

The difference between the ordinates of *CD*, and either *EF* or *GH* as the case may be, represents the heat per sec. added to or given up by the water system by virtue of its thermal capacity.

After a certain period of steady climbing the actual rate of heat dissipation will approach asymptotically to the curve *CD*, and after this has occurred the ratio of the ordinates of *AB* and *CD* gives the suitability figure for the radiator at each height. Since the normal relationship of p, σ, and θ_a to the height are such as to make the curve *CD* fall more rapidly than *AB*, it is clear that for a normally aspirated engine there will in general be an increase of suitability with height, i.e. that unless there is some reserve of heat capacity through the water starting cold, the most severe conditions will be met with at low heights.

Turning now to the question of a test in a non-standard atmosphere, when the relative temperature and density associated with a relative pressure p are θ'_a and σ'. Imagine the aeroplane flying at a height corresponding to p, and let the mean radiator temperature then be θ'_R.

The heat per sec. received from the engine will be unaffected by the air temperature, and therefore provided the climb has continued long enough for starting conditions to be eliminated

$$(\theta'_R-\theta'_a)\sigma'^n = (\theta_R-\theta_a)\sigma^n$$

and therefore the suitability figure

$$S = \frac{\theta_m-\theta_a}{\theta_R-\theta_a} = \frac{\theta_m-\theta_a}{\theta'_R-\theta'_a} \times \left(\frac{\sigma}{\sigma'}\right)^n = \frac{\theta_m-\theta_a}{\theta'_R-\theta'_a}\left(\frac{\theta'_a}{\theta_a}\right)^n.$$

How much θ'_a/θ_a may differ from unity will depend upon where the test is carried out. For temperate climates the maximum summer temperature is taken as 23° C. at the ground, with a lapse rate of very nearly 2° C. per 1,000 ft. The standard temperature being 15° C. at the ground, with the same lapse rate, θ'_a/θ_a is never likely to be more than 1·03 in temperate climates. The fraction $(\theta_m-\theta_a)/(\theta'_R-\theta'_a)$, which can be obtained from observations at any height corresponding to p, as given by the aneroid, therefore gives the suitability to within about 1·5 per cent.

Suppose, now, that a test be started with the thermometer reading 75° C. at the water inlet to the radiator. The difference between the temperature here and the true mean temperature, θ'_R, will be small compared with $\theta'_R-\theta'_a$, and it may be assumed that θ'_R is obtained direct from this thermometer reading.

Readings of θ'_R and θ'_a are taken every 1,000 ft., while θ_m and θ_a for each height can be found from tables. The water temperature will rise during the early stages of the climb (unless the radiator is *much* too large) and the suitability figure S will reach a minimum value at some height corresponding to the vertical YY in fig. 68. A minimum figure $S = 1{\cdot}05$ is generally accepted as providing a suitable margin of safety, and the height at which this minimum

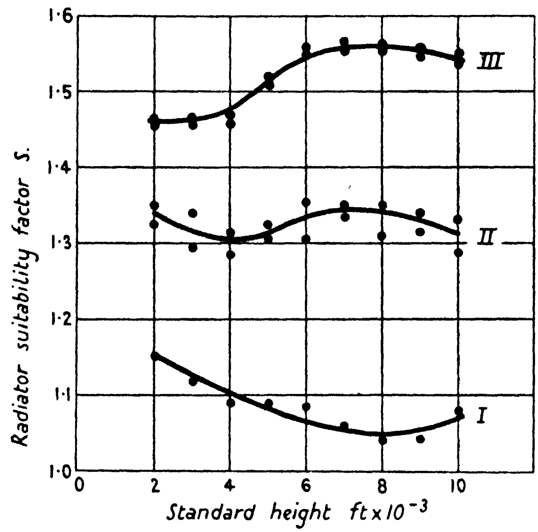

FIG. 69. Radiator suitability curves for an aeroplane with retractable radiator, the exposed areas being in the ratios 1 : 1·4 : 1·75 in the trials numbered I, II, and III.

should be reached will depend upon the type of aeroplane. The curves of fig. 69 show the variation of the suitability figure obtained during three sets of trials of the same aeroplane.[35] This was fitted with a retractable radiator projecting by an adjustable amount below the bottom surface of the fuselage, so that the quantity of water carried and the thermal capacity of the system was the same throughout, but the frontal area of the radiator exposed for cooling purposes was in the proportion of 1 : 1·4 : 1·75 in the three trials numbered I, II, and III in the figure.

With the smallest effective radiator a suitability 1·05 would have been reached at 8,000 ft. in the standard atmosphere, and the installation would not have been safe with any smaller radiator. With the greater proportion of surface exposed the suitability was unnecessarily high and the engine was overcooled; but the more important result of having too large a radiator would be the unnecessary

increase of drag. The extra radiator exposed in trial III as compared with I had the effect of reducing the top speed of the aeroplane in level flight at 5,000 ft. from 148 to 143 m.p.h.

With a radiator of the type used in these trials the rate of heat dissipation can always be adjusted to the minimum necessary, by retracting it. With a fixed radiator, however, some other means must be provided, usually in the form of shutters, to prevent overcooling during a long glide with the engine throttled down or whenever the necessary rate of heat dissipation falls below that provided by the flight conditions. When heat dissipation is controlled by shutters it is important that the bringing of these into operation should reduce, and not increase, the drag of the radiator. When the rate of cooling

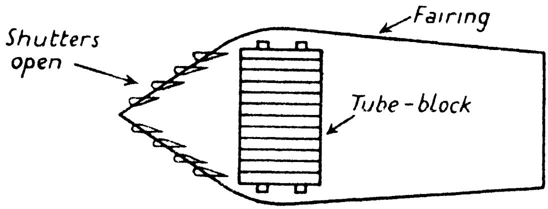

FIG. 70. Plan view of underslung radiator enclosed in fairing of which the shutters form, when closed, the streamlined nose.

is reduced by the in-drawing of a retractable radiator, the radiator drag is reduced roughly in proportion; but if the shutters are closed across the frontal surface of a fixed radiator, the reverse effect is produced, for the drag of a plane surface normal to the air-stream is greater per sq. ft. than that offered by a honeycomb radiator which allows a through passage for much of the incident air.

In order to give some further idea of the importance of radiator drag, the trials already quoted may be referred to.[35] In these the complete retraction of the radiator, until it formed no excrescence below the fuselage at all, enabled the top speed at 5,000 ft. to be increased from 148·5 m.p.h. (corresponding to the full normal radiator setting of curve I, fig. 69) to 155 m.p.h. On the other hand, the closing of the shutters across the corresponding fixed radiator caused a drop in the top speed to 147 m.p.h.

When expressed as the drag in lb. due to the radiator at an air speed of 100 ft. per sec., these figures mean a reduction from 22·8 lb. to zero for the retractable radiator, as compared with an increase from 25 to 32 lb. for the flat shuttered radiator.

A great improvement can be made in the shuttered radiator if the shutters are arranged as in fig. 70, so that when closed they form, with the fairing at the sides and rear, an approximately streamlined form.

When the same radiator block was treated in this way, instead of being closed by flat shutters, the drag at 100 ft. per sec. was reduced from 21·1 to 16·7 lb. by the closing of the shutters, instead of being increased from 25 to 32 lb. It will be noticed, too, that the drag with fully open shutters was reduced from 25 to 21·1 by the fairing, although the maximum cross-section of the structure containing the radiator had been increased by the fairing from 2·80 to 3·40 sq. ft.

ART. 50. *High-temperature liquid cooling.*

The exit temperature of the water from the cylinder-jackets of a water-cooled engine would be limited by a radiator of the normal suitability figure to about 80° C., when flying level at full throttle at heights up to 5,000 ft. The temperature difference $(\theta_K - \theta_a)$ upon which the rate of heat dissipation depends, is therefore limited to something of the order of 60° C. under summer conditions.

As can be judged from the figures given in the last article, a very substantial fraction of the useful power of an engine must be regarded as being used up in overcoming the radiator drag which is inseparable from this necessary dissipation of waste heat. The exact fraction of the total engine power required for cooling purposes will of course vary with the design and installation of the radiator, and the fuselage on which it is fitted. In the trials already quoted, in which complete removal of the normal-sized honeycomb radiator allowed the top speed at 5,000 ft. to be increased from 148·5 to 155 m.p.h., the radiator must have been responsible for about 9 per cent. of the total engine power at the lower speed.

It is clear, then, that any large increase in the available temperature difference $(\theta_K - \theta_a)$, allowing of a proportionate reduction in the necessary cooling surface and in radiator drag, holds out the possibility of an appreciable gain in performance.

In order to obtain such an increase in the available temperature difference, liquids with a higher boiling-point than water have been used as the cooling medium, of which the most important is ethylene glycol. This is a colourless liquid, rather viscous when cold, which boils at 195° C. Its chemical formula is $C_2H_4(OH)_2$, and it belongs to the class of 'dihydric' alcohols, containing as it does in its molecule two (OH) groups as compared with the one of these in ethyl alcohol $C_2H_5(OH)$.

A liquid with the necessary properties for a high-temperature cooling medium is not easy to find, for besides having a high boiling-point, it must be chemically stable when maintained for long periods at the working temperature, and must not be so viscous at lower temperatures that circulation by the pump is hindered before it has

been thoroughly heated up. An exhaustive search among possible organic liquids has failed to produce any competitor to ethylene glycol except a mixture of ordinary glycerine and water in the proportion of about 9 : 1. Pure glycerine would be too viscous at low temperatures, and the mixture has the serious drawback that during prolonged use at high temperatures the water is driven off and the residue becomes progressively more viscous.

Ethylene glycol, at 25° C., is about seventeen times more viscous than water, but the difference is not sufficient to affect circulation seriously, and at 80° C. the rate of delivery by the common type of centrifugal pump is just about the same, measured in gallons, for the two liquids. At higher temperatures the rate of delivery of water falls off rapidly, until at 95° C. it is only about 75 per cent. of the value at 80° C., the reason for this being the rapid expansion of water near its boiling-point and the liability to vapour formation on the suction side of the pump. With ethylene glycol, on the other hand, the rate of delivery increases substantially for a constant pump speed, and at 140° C. it is 20 per cent. higher than at 80° C. This is fortunate, for the specific heat of ethylene glycol is only 0·62 times that of water, and the higher rate of circulation, combined with a specific gravity 10 per cent. greater than water, is able to compensate for the low specific heat; so that the same pump can be used for the two liquids.

Under the normal conditions of water cooling as applied to aero-engines, the rise of temperature of the water between inlet and outlet to the engine may be taken as about 10° C., with a corresponding fall across the radiator. With ethylene glycol, operating with an upper temperature of 150° C., the rise of temperature in passing through the engine would be increased in the ratio of the specific heats and reduced in proportion to the specific gravities and the rates of delivery of the two liquids, assuming the same pump to be used, and would therefore be approximately

$$10 \times \frac{1}{0{\cdot}62} \times \frac{1}{1{\cdot}1} \times \frac{1}{1{\cdot}20} = 12{\cdot}2^\circ \text{ C.}$$

if the same amount of heat be assumed to be received from the cylinders per min. In fact the heat received will be somewhat less by reason of the higher temperature of the cylinder walls, and we should expect therefore about the same rise of temperature in the cooling fluid whether an engine employs water at 80°–90° C. or ethylene glycol at 140°–150° C.

A full account has been given by Frank[43] of experiments in which glycol cooling was applied to two types of American water-cooled

engines, the Curtiss V. 1570 and Curtiss D. 12. These are each 12-cylinder V-type engines of which the leading particulars are given in table 39.

TABLE 39

Particulars of Curtiss engines, V. 1570 *and D.* 12, *each of* 12-*cylinder V-type.*

	V. 1570	D. 12
Bore × stroke	$5\frac{1}{8}$ in. × $6\frac{1}{4}$ in.	$4\frac{1}{2}$ in. × 6 in.
Normal full speed	2,100	2,300
Compression ratio	5·8 to 1	5·6 to 1
Angle of *V*	60°	60°
Piston displacement	1,569 cu. in.	1,160 cu. in.
Rated full power	525 h.p.	435 h.p.

The results of this work must be considered, so far as the engine is concerned, from two points of view. The results to be expected from high-temperature liquid cooling when installed in an aeroplane constitutes yet a third aspect, but for the present attention will be confined to the engine. Here the points of view are (*a*) the effect of the high jacket temperature on the thermodynamics of the engine and the consequent change of power output, and (*b*) the effect upon the engine, as a mechanism, of the combined chemical nature and high temperature of the cooling liquid.

The high jacket temperature may be expected to affect the power output adversely through a slight reduction of volumetric efficiency and, with some fuels, more seriously because of an earlier promotion of detonation. On the other hand the higher temperatures will reduce the friction horse-power, and may raise the mechanical efficiency by an amount sufficient, at high altitudes, to produce a nett gain of B.H.P. (see pp. 328–9).

In fig. 71 are shown the full-throttle B.M.E.P. and fuel consumption curves for the D. 12 engine with water and glycol cooling.

In order to make full throttle running possible with the glycol cooling the proportion of benzol in the fuel was increased from 20 to 40 per cent. This would lower the calorific value per lb. by 2 per cent., so that under maximum power conditions the brake thermal efficiency was identical, within the limits of accuracy of the test. When running at the most economical fuel-air ratio, however, it will be seen that the fuel consumption at 2,300 r.p.m. was lowered from 0·495 to 0·48 lb. per B.H.P. hour by the high-temperature cooling. Friction h.p. was not measured up to the higher temperature, but by extrapolation it may be said that the mechanical efficiency would not have been increased by more than 1 per cent. at the outside, and

probably less. The reduction of 3 per cent. in the fuel consumption, therefore, which represents an increase of 5 per cent. in overall thermal efficiency, must have been due to better carburation and a more uniform mixture of the fuel and air in all the cylinders on account of the higher temperatures of the cylinders and the induction manifold.

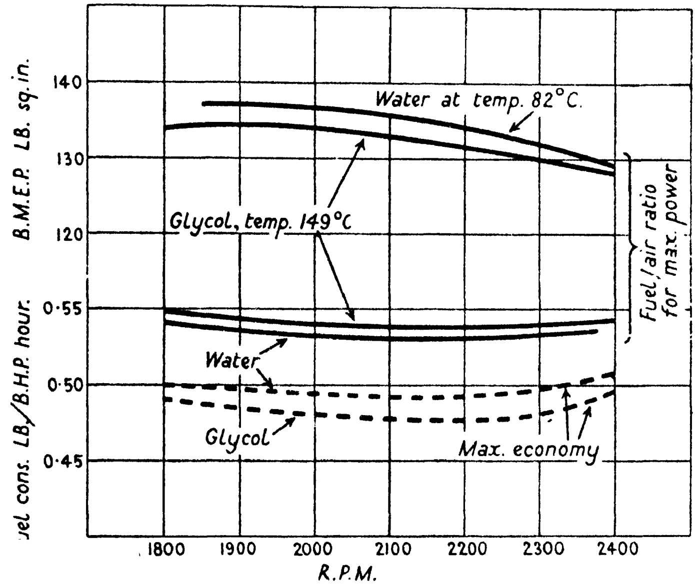

FIG. 71. Curves of B.M.E.P. and fuel consumption for Curtiss D. 12 engine under normal and under high-temperature cooling conditions.

The measured B.H.P.s at the normal speed were 436·5 and 432·5, so that the volumetric efficiency was lowered by the higher jacket temperature by about 1 per cent. The oil consumption was normal during both sets of trials, at 0·013 lb. per B.H.P. hour at full speed.

During the tests, the temperatures at points within the thickness of the cylinder walls were measured for both engines, and to understand the significance of these measurements, as well as certain troubles which were met with, it is necessary to observe the different types of cylinder construction employed in each engine.

In figs. 1 and 2 on p. 8, there were shown cross-sections of the two cylinders, side by side, with the locations of the thermocouples marked, in fig. 1 at *C* and *D* and in fig. 2 at *A* and *B*. The important thing to notice is that in fig. 2 the steel liner *C* was an open-ended tube, so that the thermocouple *A* was in the aluminium alloy head,

and separated only by ¼ in. of the metal from the cylinder gases. In the D.12 engine shown in fig. 1 the inner surface of the cylinder head *A* was of steel, integral with the barrel, and the thermocouple *C* was in that case embedded in the steel liner. Higher temperatures would therefore be anticipated, because the heat-flow to the cooling fluid had in that case to traverse the steel-to-aluminium joint beyond the thermocouple.

TABLE 40

Cylinder head and barrel temperatures with water and glycol as cooling fluids. Temperatures given are the averages of the 12 cylinders.

	Thermocouple	*Cooling liquid and exit temperature*		
		Water 82° C.	*Glycol* 82° C.	*Glycol* 149° C.
Engine V.1570	*A* (cylinder head)	154 (163–149)	191 (197–186)	222 (230–220)
	B (cylinder barrel)	88	101	157
Engine D.12.	*C* (cylinder head)	192 (205–182)	236 (249–221)	265 (282–257)
	D (cylinder barrel)	86	98	159

In table 40 the temperature observations are summarized. The figures given are in each case the average for all 12 cylinders. Among the barrel temperatures the variation was unimportant, and on the V. 1570 engine, without the steel liner, the extreme variation of head temperature was no more than 14° C. On the D.12 engine it was greater, and, as would be expected, it was most marked when using glycol cooling at the lower temperature, i.e. when the viscosity of the cooling medium was high. The actual ranges of temperature variation between the cylinders are shown in brackets below the mean values of the thermocouple readings.

Other points to notice about the figures of table 40 are that, while the barrel temperatures do not differ appreciably from one engine to the other, the steel liner in the D.12 engine has put up the thermocouple temperature, at ¼ in. from the cylinder-head surface, by about 40° C.; and further, that as between water and glycol at the same temperature, 82° C., the cylinder-head temperature in each engine was 40° C. higher with the glycol. This fact is important, for although no one would suggest using glycol at the low temperature, it shows how a high viscosity affects the coefficient of heat

transfer from metal to liquid and vice versa, and therefore that it must not be assumed that the comparative rates of heat dissipation from a glycol and a water radiator can be calculated simply on the basis of a change in the available temperature difference $(\theta_R - \theta_a)$. It is of interest that the difference between the average head temperature and the exit temperature of the fluid is almost exactly the same for water at 82° C. and for glycol at 149° C. The differences are 72° and 73° in the first engine and 110° and 116° in the second.

A maximum temperature of 150° C. at entry to the radiator would provide an increase in $(\theta_R - \theta_a)$ of just about 100 per cent. Data are lacking for an estimate to be made as to what reduction of radiator surface this would allow. In the American trials the experimental aeroplane with glycol cooling was flown with only 30 per cent. of the radiator surface which would have been required for water. The operation of the engine and installation is said to have been very satisfactory, but full information is not given as to the tests carried out.

Glycol cooling may be said to possess a margin of spare cooling capacity, in so far as an installation designed to work at a maximum temperature of 150° C. could on occasion be run up to higher temperatures, with an ever-increasing rate of heat dissipation, and without serious danger from boiling away of the liquid. There might be additional trouble from engine details, and this aspect of the use of glycol cooling must now be shortly referred to. With higher cylinder temperatures, all effects of differential expansion as between steel and aluminium will be exaggerated. Piston clearances must be increased, and where the cylinder-jacket joints are affected, leaks must be expected. Trouble of this kind was experienced in the American engines at the joint between the aluminium jacket and steel barrel at the lower ends of the cylinders, and a special form of compound packing had to be adopted. Leakage troubles are accentuated through the ability of glycol to creep over surfaces and through the most minute cracks, just as the familiar paraffin does. During the trials with the D.12 engine the glycol was found to be passing right up past the screwed connexion between the steel liner and the aluminium head, and to be entering the cylinder through the inlet valves (see fig. 1).

These are incidental troubles, to be got over by proper attention to the details of design and development; and as for the higher than normal temperatures, they are anyway not so high as in the air-cooled engine. Whether or not the margin of improved performance, obtained through the reduced size of radiator needed with high-

temperature liquid cooling, is sufficient to justify the development, is a question of policy with which we are not here concerned. It was shown in the last article that the complete removal of the normal water radiator allowed an increase of the maximum level speed of a fast aeroplane from 148·5 to 155 m.p.h. at 5,000 ft. A gain of 4 m.p.h. under the same conditions is therefore likely to be the most that glycol cooling could have given.

IX

THE FUEL-AIR MIXTURE SUPPLY. CARBURETTORS.

ART. 51. *The fuel-air ratio and its relation to power output.*

It is well known that there is a certain 'correct' ratio of fuel to air which gives just complete combustion, and there will be a certain optimum ratio appropriate to the circumstances, according to whether an engine is adjusted to give its maximum power or to work with the greatest possible economy in fuel per B.H.P.

In a single-cylinder test engine at ground-level the optimum ratio varies between about 15 per cent. rich for maximum power to 15 per cent. weak for maximum economy. In a multi-cylinder aero-engine it is not normally possible to work with an *average* fuel-air ratio weaker than the correct mixture, and for full throttle operation it is desirable as a rule to provide a mixture rather more than 15 per cent. rich. The need for some over-enrichment, superfluous so far as power output is concerned, arises from the fact that the dangers of overheating are lessened thereby, partly by reason of the extra cooling from the evaporation of the surplus fuel, and partly because the explosion temperatures are reduced by the higher volumetric heat of the cylinder gases which result from a rich mixture.

It has already been emphasized in Chapter III that the critical problems in a high-power engine are associated with the very severe heat-flow in pistons, valves, and cylinder-heads; and most engines are designed so as to rely upon the help afforded by an over-rich mixture during the limited periods when they are likely to be operating with fully open throttle.

It will be seen, therefore, that in the normal multi-cylinder aero-engine there is a shift of the working range of mixture strengths towards the rich end, as compared with the thermodynamic needs of the cycle demonstrated by a test engine; and this tendency towards enrichment is likely, if anything, to become more marked at high altitudes where the air temperature and density are low.

It is the function of the carburettor to ensure that each pound of air, upon entry into the induction system, shall have associated with it the proper weight of petrol to give the mixture required by the engine, according to its changing conditions of working. While it is no part of the carburettor's business to see to the evaporation of the petrol, it is possible by good design so to arrange that the petrol leaves it in a fine spray, and so in a condition to evaporate more rapidly. How completely and rapidly it will do so depends, further,

upon the temperature of the air supply, the volatility of the petrol, and the amount of heat supplied to the induction system.

The ideal of the carburettor is that the petrol supply should automatically be adjusted to the air supply under all conditions of height, throttle setting, and engine speed; and that this adjustment should be achieved purely by the design of the passages which control the flow of the air and the fuel, without any moving parts. Such a thing is clearly impossible so long as there remains any element of choice as to whether the engine shall work with a rich or a lean mixture for the same speed and throttle setting, that is, for the same flow of air through the carburettor. It happens, however, that in an aeroplane there is less need to retain this element of choice than might be supposed. Engines must be allowed an over-rich mixture while operating at full throttle, but no engine would normally be expected to run for long periods at full throttle, and while it is doing so we may be content to sacrifice economy. It is while cruising in level flight at something well below the maximum forward speed that fuel economy is most important. In these circumstances the engine would be throttled to give only $\frac{2}{3}$ to $\frac{3}{4}$ of its maximum power, and the reduced air-flow through the carburettor can be accompanied by a more than proportionate reduction of the fuel, so that a leaner mixture appropriate to the cruising power of the engine is supplied, and the economy of the engine thereby improved.

In level flight there is for each forward speed, and corresponding r.p.m. of the airscrew, one definite, necessary value of the B.M.E.P. of the engine, and therefore of the throttle position and of the flow of air through the carburettor.* The requirements both of fuel economy and of full power will therefore be met if the carburettor is capable of maintaining an economical mixture for all throttle openings which correspond to B.M.E.P.s below about 85 per cent. of the full-throttle value, and if at higher powers there is a gradual enrichment which may increase the rate of fuel consumption per B.H.P. by 15–20 per cent. while operating at full throttle.

It must be noted that a variation of engine r.p.m. and a variation of the throttle position at constant r.p.m. might each have the same effect upon the air-flow through the carburettor. It follows that a carburettor in which the petrol flow was automatically controlled purely by the air-flow, must needs give the same rich mixture at high speeds with partially closed throttle as it would be required to give with fully open throttle at the lower speeds experienced while climbing. The point is not an important one, however, because the only conditions in which a high engine speed could be associated with

* For a fuller discussion of this important point, see Chapters XII and XIV.

a partially closed throttle would be during a dive, and this is not a condition likely to last long nor one in which fuel economy would be of any importance.

Corresponding to each value of the fuel-air ratio, at any given speed and power, an engine will show a certain rate of fuel consumption per B.H.P. hour, and the most practical form in which to give quantitative expression to the proper behaviour of a carburettor in relation to the throttle position, is to show how the rate of fuel consumption per B.H.P. hour for an average engine should be corre-

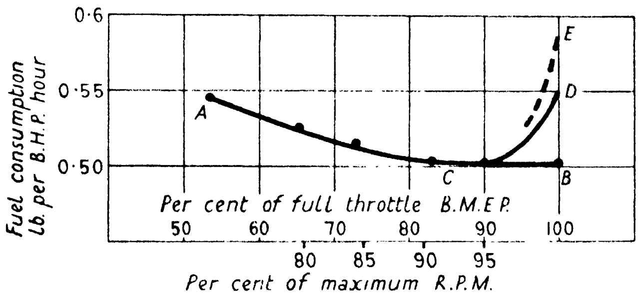

FIG. 72. Variation of fuel consumption per B.H.P. hour with throttle opening, and engine r.p.m. in level flight.

lated with the B.M.E.P., or with the percentage of the full-throttle B.M.E.P.

An aeroplane might cruise with its engine throttled to give anything between 50 and 85 per cent. of its full B.M.E.P. at ground-level, and this is the range of power, therefore, over which the carburettor should provide an economical fuel-air ratio, allowing the mixture to 'richen up' towards full throttle.

The curve *BCA* in fig. 72 shows the form of consumption curve at different B.M.E.P.s, from full throttle downwards, which would be given by an engine of average mechanical efficiency in level flight if supplied throughout the full range of power with a constant fuel-air ratio suitable for economical flying. The alternative portion, *CD*, shows how the fuel consumption should be allowed to increase towards full throttle from its minimum value of 0·5 lb. per B.H.P. hour up to 0·55 lb. at the full B.M.E.P., and the dotted curve *CE* illustrates the still greater degree of enrichment which is normally allowed for an air-cooled engine.

It will be seen that a scale of engine speeds is given, expressed as a percentage of the maximum r.p.m. The relation between the engine torque, or B.M.E.P., and the r.p.m. in level flight will depend upon what is the maximum r.p.m. allowed by the airscrew at

full throttle. The normal arrangement is to employ an airscrew which allows the engine to run at its maximum permissible r.p.m. when at full throttle in level flight, and from that condition downwards the power will vary roughly as the cube of the r.p.m. and therefore the B.M.E.P. (which is proportional to the torque) roughly as the $(\text{r.p.m.})^2$.

Throughout the whole length of the full-line curve *BCA* the carburettor would be supplying an approximately constant fuel-air ratio, but there is a rise in the fuel consumption per B.H.P. hour from full throttle downwards by reason of the increasing proportion of the total power which is absorbed by the friction and pumping losses. During the same variation of B.M.E.P. in a test made at constant r.p.m. the rise of fuel consumption per B.H.P. at low powers would be substantially greater. It is an important point to remember that when an engine is throttled down in level flight the concomitant fall in speed allows the engine to maintain, within limits, it mechanical efficiency.

The curves *ACD* and *ACE* are not intended to define the best that can be achieved in the way of fuel economy: they are given merely to illustrate in a quantitative way how the specific fuel consumption of an engine should vary in level flight. It should be possible slightly to improve upon the minimum figure of 0·5 lb. per B.H.P. hour, at the point *C*, with a high compression-ratio engine and with special mixture control devices for obtaining an economical mixture. The curves of the figure illustrate the characteristics of the unaided carburettor without special precautions for obtaining a high economy, and they would apply to a normally aspirated engine of about 6 : 1 compression ratio.

It is simpler, and advisable for other reasons, to provide the enrichment *CD* towards full throttle by some mechanical device which depends, not upon the air-flow through the carburettor, but upon the position of the throttle lever. The carburettor is still automatic in the sense that no separate manual adjustment is called for, but a mechanical moving part has been introduced to supplement the automatic adjustment provided by the design of the fuel and air passages.

With this special provision for the full-throttle enrichment, the problem of carburettor design becomes that of providing such an arrangement of the passages for air and fuel that a constant economical mixture shall be maintained under all conditions of air-flow. There are certain other practical requirements, e.g. for obtaining satisfactory 'idling' and rapid acceleration; but these lie outside the main principles upon which the working of the carburettor

depends. They will be described, along with a typical device for obtaining the full-throttle enrichment, at a later stage; for the present the discussion will be limited to the principles which must be complied with by any design which aims at providing a constant mixture strength with varying rates of air-flow.

If we leave out of account for the present the necessary compensation for a change of altitude, then the ideal of the entirely automatic carburettor with no moving parts becomes a perfectly practicable, although not an easy, one to achieve.

The problem, stated in its simplest terms, is that of designing an instrument in which there are two passages, one carrying a stream of petrol and the other a stream of air, which are so shaped that as the mass-flow of air varies in the one, the mass-flow of petrol in the other is altered in exact proportion.

Apart from its relation to the fixed dimensions of the passages, the mass-flow of air and petrol will be related to the pressure difference between the ends of each passage, and will also depend upon the viscosity and density of the fluids flowing. Air being a compressible fluid, and petrol incompressible, the relation between the rate of flow and the pressure difference will, as already mentioned, be radically different for the two. Furthermore the vast difference of viscosity and density between the two fluids means that their behaviour in relation to the shape of the passages will be widely different. And finally their differences of density will mean that the response to a sudden change of the pressure difference which is causing the flow will be dissimilar. Owing to its greater inertia an increase of flow in the petrol will lag behind the sudden increase of air-flow which follows an increase of the throttle-opening. It has been suggested that this difference of density is responsible for the temporary weakening of the fuel-air mixture, and failure of the engine to accelerate, which is sometimes experienced when the throttle is quickly opened. Calculation shows, however, that if an engine is idling and the throttle is suddenly thrown fully open, the time for the fuel-flow to reach 95 per cent. of its final value is never likely to amount to more than ½ sec., and this delay would have no appreciable affect on the engine.

The real causes of bad acceleration are probably, in order of importance:

(1) That when the throttle is quickly opened there is momentarily insufficient suction to operate the slow-running jet, while at the same time the rate of air-flow, until the engine has speeded up, is not sufficient to produce a suitable mixture from the main jet without a contribution from the other.

(2) That part of the petrol from the main jet is projected on to the walls of the choke tube, and owing to the insufficient velocity of the air through it the petrol fails to get carried forward into the induction system.

(3) That during idling the walls of the induction system may have become chilled by evaporation of the fuel under a partial vacuum, and

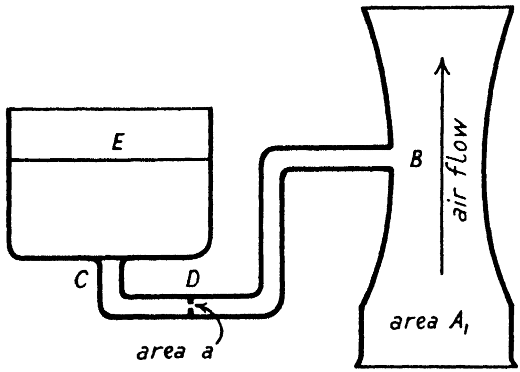

FIG. 73. The simple carburettor with submerged orifice for metering the fuel flow.

when it is suddenly filled with air at more than twice the pressure the walls are temporarily too cold to act as effective evaporating surfaces.

'Good acceleration', that is, an immediate and steady response to the opening of the throttle, is a vital necessity in an aeroplane engine, for upon it the question of safety or disaster may often depend; and special devices for ensuring an ample supply of petrol to the induction system, to be referred to in art. 55, must therefore be provided. In the next article the theory of the flow of petrol and air under steady conditions will be given; at first for each treated separately, and then combined so as to set forth the necessary flow characteristics if a constant petrol-air ratio is to be maintained. In a later article the questions of pulsating flow, and of sudden changes of flow, will be discussed.

ART. 52. *Air and petrol flow under steady conditions.*

Consider the simplest possible form of carburettor, as outlined in fig. 73, consisting of an air passage of Venturi form with a minimum cross-sectional area at B, and a fuel passage with its exit at B, at the same level as that maintained by the free surface of the petrol in the float chamber at E.

As air flows through the Venturi a pressure difference p_1 is

established between the throat and the outside air. The float chamber being subject to atmospheric pressure, petrol flows to the jet at B under the presure difference established by the air-flow. It will be observed that the petrol passage CB is shown as of comparatively large size with a small orifice D which is not near either end. The rate of flow of petrol will be entirely controlled by the flow through this orifice, which means that there will be a pressure difference between the two sides of the orifice equal to the difference between the ends of the passage, the petrol in the large-bore passage before and after the orifice behaving as though stationary. For reasons to be given later the use of a submerged orifice for metering the petrol is found to be much more satisfactory than to have the metering orifice, for example, at the exit end B.

Now let

P and T be the pressure and abs. temperature of the air at entry to the carburettor,

p_1 be the pressure difference between the point B and the outside air,

p_2 be the pressure difference between the ends of the fuel passage, (in the simple carburettor $p_2 = p_1$, but this will not be assumed at present),

A_2 and $A_1 = kA_2$ be the areas of throat and entry to the carburettor,

ρ_2 and ρ_1 be the air density at throat and entry,

σ „ fuel density,

a „ area of the fuel orifice.

If air and petrol were both to behave as inviscid and incompressible fluids, then the mass-flow of each could be derived from Bernoulli's equation and expressed as follows:

$$W_a = \text{constant} \times (p_1\rho_1)^{\frac{1}{2}},$$

$$W_p = \text{constant} \times (p_2\sigma)^{\frac{1}{2}}.$$

The flows would be proportional in the simple carburettor, and a constant mixture strength would be maintained, apart from changes of σ with change of height. Air, however, is not incompressible, and petrol is by no means inviscid, and the result is that the relationship between mass-flow and pressure difference can in neither case be expressed in the simple form given above.

Taking first the case of the petrol flow through an orifice, the fluid here is incompressible, but its viscosity will affect the rate of flow to a degree which depends very much upon the form of the orifice and upon the pressure difference.

We may write

$$W_p = 5{\cdot}28\, Ca(p_2\sigma)^{\frac{1}{2}} \tag{35}$$

if W_p is expressed in lb. per sec.
p_2 " " lb. per sq. in.
a " " sq. in.
σ is the S.G. of the fuel relative to water at 15° C.

In this equation C is a coefficient which allows for the variations of flow introduced as a result of viscosity in the liquid. The value of C

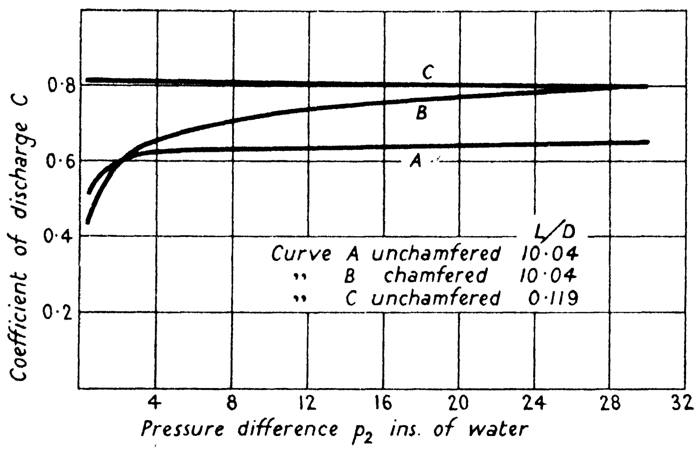

FIG. 74. The effect of chamfering the ends of the orifice upon the coefficient C. Submerged orifice; water-flowing; temp. 20° C.

will cover all losses due to skin friction as well as those associated with changes of cross-section in the passage. It will vary with p_2 and with changes in the ratio of length to diameter (L/D) of the fuel orifice, as well as with changes of temperature (and therefore viscosity) in the fuel. The value of C will decrease for a fall in temperature (increase of viscosity) to a degree which depends upon the type of orifice.

The ways in which C varies according to the type of orifice and its L/D ratio are shown in figs. 74 and 75.[44] In these diagrams the values of p_2 are expressed in inches of water, as being more convenient figures to deal with. The full range of p_2 met with in carburettor practice may be taken as about 25 in. of water.

The first point to notice in fig. 74, in connexion with the shape of the orifice, is the difference of behaviour according to whether the ends of the orifice are left square or chamfered. Curves A and B give a direct comparison, for an orifice in which $L = 0{\cdot}4$ in. and $D = 0{\cdot}04$ in., between the unchamfered and chamfered condition.

The degree to which the edge was chamfered away made very little difference, once the sharp edge had been removed. It is clear that while chamfering allows the coefficient to reach higher values, it also makes it far less constant in value over a range of pressure differences.

The reason is that constancy of the coefficient corresponds to flow according to the Bernoulli equation, in which pressure differences are proportional to v^2, and this only holds good for a non-viscid liquid or for fully turbulent flow in a liquid which has viscosity. Under conditions of streamline flow in a viscous liquid the pressure difference would be proportional to the first power of the velocity.

According to the degree to which flow through the orifice partakes of the nature of viscous streamline flow, so will departure from the assumptions of the Bernoulli equation show up as a falling off in the value of the coefficient. It is for this reason that the falling off shown by curves A and B is most marked at small values of p_2, when the flow approaches more and more to one of a streamline character.

The only form of orifice through which the flow is of a truly turbulent character is a sharp-edged hole in a thin plate, and for such an orifice the almost perfect constancy of the coefficient C at all values of p_2 is shown by the line C in fig. 74. Even in the unchamfered orifice of curve A the large L/D ratio of 10 introduces marked viscosity effects into the flow and prevents constancy of the coefficient. In the orifice of curve C, L/D was only 0·1 while $D = 0{\cdot}04$ in., so that the orifice had the form of a hole in a thin plate, and even at the smallest value of p_2 the coefficient had a value equal to the highest reached by orifice B.

The constancy of the coefficient exhibited by a sharp-edged 'thin plate' type of orifice leads us to the important conclusion that if the petrol metering orifice in a carburettor were of this type the rate of flow through it would be strictly proportional to $(p_2)^{\frac{1}{2}}$ and, apart from the effect of air compressibility, which is quite small, the perfectly simple single-jet carburettor of fig. 73 would yield a constant mixture strength at all throttle openings. In other words, all the complications of the compensated carburettor as applied to the motor-car—i.e. excluding compensation for altitude—would be unnecessary if viscosity effects in the petrol could be eliminated by using the right kind of orifice, and if certain practical requirements did not make it necessary for the working level of the petrol in the float chamber to be below the level of the point B in the simple carburettor of fig. 73.

There is a purely practical, but none the less strong, reason for not using the ideal thin plate orifice. It is because the sharp edge of the orifice has to be very carefully made, to avoid the least trace of burr

or irregularity. Very small blemishes at the edge may produce a large effect upon the coefficient, and as a manufacturing proposition it is found safer to use chamfered jets, for which no more than reasonable care is necessary, and to compensate for the viscosity effects introduced thereby.

Having once accepted the orifice with chamfered ends, variation of the coefficient is to be expected according to its L/D ratio, to the

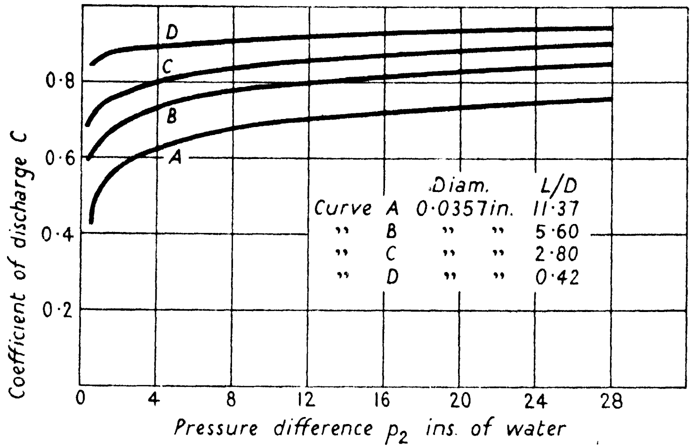

FIG. 75. Effect on C of a change in L/D with D constant. Water-flowing; temp. 24° C.; submerged orifice; chamfered ends.

temperature of the petrol flowing, and to the pressure difference. The curves of fig. 75 show how the coefficient varies with the L/D ratio and pressure difference for a submerged and chamfered orifice, with distilled water flowing, and fig. 76 shows the effect of a change of temperature when petrol is flowing through an orifice of nearly similar proportions to those of A in fig. 75.

It is clear from fig. 75 that a small L/D ratio has the advantage of giving a higher coefficient, and one much less affected by changes in the pressure difference. In such an orifice the effect of temperature would be much less marked than it was for the orifice of fig. 76, with an L/D ratio of 11·83. There a temperature rise of 20° C. produced an increase of about 5 per cent. in the coefficient, but in a short orifice such as D of fig. 75 that temperature difference would produce an effect scarcely noticeable.

It must be observed, in connexion with fig. 75, that the coefficient would be affected by a change of orifice diameter, apart from any change in the L/D ratio. In making calculations of the necessary jet sizes for a carburettor, the values of the coefficient would have

to be chosen appropriate to the L/D ratio and to the approximate range of diameters which it was intended to use.

Having now examined to what degree, and for what reasons, the mass-flow of petrol through the metering orifice will not be proportional to $(p_2)^{\frac{1}{2}}$, it remains to examine the relationship between the mass-flow of air through the Venturi and the pressure drop p_1, and how p_2 is in practice related to p_1.

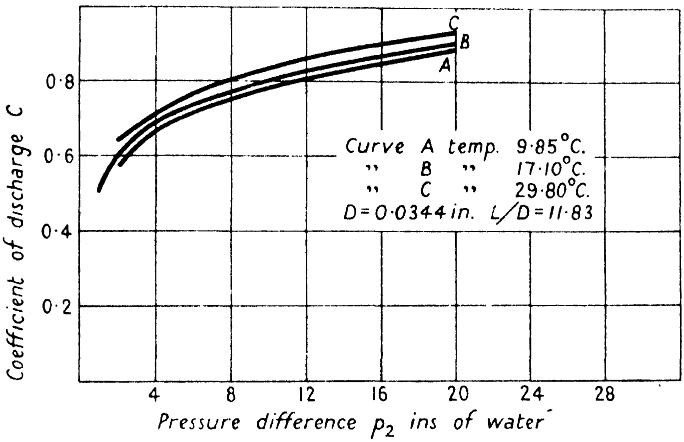

FIG. 76. Effect on C of a change in fluid temperature. Petrol-flowing; submerged orifice; chamfered ends.

When a compressible fluid flows adiabatically and without turbulence through a Venturi tube its behaviour can be defined in terms of three equations:

(1) the energy equation,

$$\tfrac{1}{2}(v_2^2-v_1^2) = -\int_1^2 v\,dp; \tag{36}$$

(2) the characteristic equation for an adiabatic change,

$$PV^{\gamma} = \text{constant}; \tag{37}$$

(3) the continuity equation

$$\rho v A = \text{constant}. \tag{38}$$

From these three equations expressions for the velocity at the throat, and the mass-flow per unit area of cross-section at that point can be derived as follows:

$$\tfrac{1}{2}(v_2^2-v_1^2) = \frac{\gamma P}{(\gamma-1)\rho_1}\left[1-\left(\frac{P-p_1}{P}\right)^{(\gamma-1)/\gamma}\right] \tag{39}$$

and

$$W_a = \left(\frac{2\gamma P\rho_1}{\gamma-1}\right)^{\frac{1}{2}}\left[\left(\frac{P-p_1}{P}\right)^{2/\gamma}-\left(\frac{P-p_1}{P}\right)^{(\gamma+1)/\gamma}\right]^{\frac{1}{2}}. \tag{40}$$

Proofs of these formulae are given in text-books of thermodynamics, and need not be repeated here.

It is clear that the relationship between W_a and p_1 is radically different from that between W_p and p_2, even when viscosity effects are neglected, and it is important to estimate the magnitude of this fundamental discrepancy between the two types of flow which arises from the compressibility of the air. This we will now proceed to do.

In carburettors the ratio p_1/P is never greater than about $\frac{1}{15}$, and by expanding the above expression and neglecting powers of p_1/P higher than the first, we obtain the expression for mass-flow in the form

$$W_a = \left[\frac{2p_1\rho_1\left(1-\dfrac{3}{2\gamma}\dfrac{p_1}{P}\right)}{1-\dfrac{1}{k^2}\left(1-\dfrac{2}{\gamma}\dfrac{p_1}{P}\right)}\right]^{\frac{1}{2}},$$

in which

$$k = \frac{A_1}{A_2} = \frac{v_2\rho_2}{v_1\rho_1}.$$

Now γ for air is 1·408, so that $3/2\gamma$ is approximately unity, and in the denominator the expression $2p_1/k^2\gamma P$ will be small compared with unity. The expression for the mass-flow of air per unit area of the throat therefore reduces to the simple form

$$W_a = \left[\frac{2p_1(P-p_1)}{RT(1-1/k^2)}\right]^{\frac{1}{2}}, \tag{41}$$

in which R is the gas constant.

Combining, now, the expressions for the mass-flow of air and of petrol, we may write down an expression for the fuel-air ratio, as follows:

$$\mu = \frac{W_p a}{W_a A_2} = \text{constant} \times \frac{a(p_2\sigma)^{\frac{1}{2}}}{A_2\left[\dfrac{2p_1(P-p_1)}{RT(1-1/k^2)}\right]^{\frac{1}{2}}}$$

$$= \text{constant}\,\frac{a}{A_2}(1-1/k^2)^{\frac{1}{2}}\left[\frac{RTp_2\sigma}{(P-p_1)p_1}\right]^{\frac{1}{2}}. \tag{42}$$

This expression* shows in a concise form what the correction for the compressibility of the air amounts to, for, if it had been incompressible, the expression in the square bracket would have been, simply,

$$\frac{p_2\sigma}{p_1\rho_1},$$

* For which I am indebted to Dr. A. A. Griffith.

and it is at once clear that in order that p_1 may be a measure of the air-flow it must be associated, not with the density ρ_1 of the air at entry, but with the density

$$\frac{P-p_1}{RT},$$

which is the density of air at a pressure equal to that at the throat, but at the external air temperature. In order to form an idea of the magnitude of the correction, suppose the depression at the carburettor throat amounted to 2 in. of mercury, corresponding at ground-level to an air velocity of about 350 ft. per sec., then the density in question would be $0{\cdot}933\rho_1$ and the effect of compressibility on the fuel-air ratio would be nearly 3·5 per cent. This, however, may be taken as an outside figure, for air velocities at the throat would seldom exceed 250 ft. per sec., which corresponds at ground-level to a depression of 1·05 in. of mercury and to a compressibility correction of only about 1·5 per cent. At high altitudes it would be greater, but not much; thus at 22,000 ft. where the density in the standard atmosphere is halved, and the pressure is 0·422 of its ground-level value, the correction would amount to 2 per cent. at the same engine speed and air velocity. It will, of course, always be in the direction of enrichment of the fuel-air mixture.

If ρ' be written for the density of air at the carburettor throat pressure and external air temperature, so that

$$\rho' = \frac{P-p_1}{RT},$$

and if, by some device in the carburettor, p_2 could always be adjusted so that

$$(p_2\sigma)^{\frac{1}{2}} = \frac{A_2}{a}(p_1\rho')^{\frac{1}{2}},$$

then, apart from the effects of viscosity, this would provide compensation for all conditions, including change of altitude; for the treatment has not been dependent upon any particular values of P, T, and ρ'.

We have seen, however, that viscosity effects at the fuel orifice cannot in practice be eliminated, and there is another practical reason, already briefly alluded to, which prevents the ratio of p_2 to p_1, and therefore the fuel-air mixture, from remaining constant when the throttle is opened and p_1 increases with the mass-flow of the air. In the simple carburettor of fig. 73 it was assumed that the free surface of the petrol in the float chamber was at the same level as the point B in the choke tube. In practice it is necessary for the working level

in the float chamber to be $\frac{1}{2}$ to 1 in. below the outlet at the choke, for even when not working the level must be sufficiently below the outlet to prevent flooding of the jets, and between this and the working condition there must be a drop of $\frac{1}{2}$ in. or so in the float-chamber level. This follows from the fact that while working the needle valve is off its seat, and while standing it must be held sufficiently tightly shut to withstand a substantial head of petrol in the tanks.

The result is that p_2 is proportional to $p_1 - g\sigma h$ instead of to p_1, and at small throttle openings, when p_1 may be no more than 2 or 3 in. of petrol, it is clear that the difference of level h may be responsible for a rapid falling off in the petrol-flow, more important than the reduction of the discharge coefficient of the jet, shown in fig. 75.

It is necessary, also, to introduce a discharge coefficient into the expression for the mass-flow of the air, just as for the fuel, in order to allow for the effect of changes in the direction of the air passage and the obstructions introduced, in the form of fuel jets and their supports, for in the foregoing treatment the air-flow was assumed to be uninterrupted, and through a Venturi of suitable shape.

The result of these inevitable deviations from theory, due to differences of level and to the variability of the discharge coefficients for petrol and air, is to force the designer to a somewhat *ad hoc* treatment of the problem of compensation by employing devices of which the correct performance can be checked by experiment. In the commonly used 'pressure balance' system of mixture control, for example, a hand-operated lever adjusts the relation between p_2 and p_1 according to the altitude and whether or not an economical mixture is required.

Carburettors differ widely in their forms of introductory air passage, in the Venturi shape, and in the design of jets for introducing the fuel into the air-stream; but such tests as are available[44] indicate that the value of the discharge coefficient varies only between fairly narrow limits. As an expression for the actual mass-flow of air through a carburettor, we may write

$$W_a = 14{\cdot}9\, C A_2 (P\rho_1)^{\frac{1}{2}} \left[\left(\frac{P - p_1}{P} \right)^{1{\cdot}42} - \left(\frac{P - p_1}{P} \right)^{1{\cdot}71} \right]^{\frac{1}{2}}, \qquad (43)$$

where W_a is expressed in lb. per sec.

P	,,	,,	lb. per sq. in.
A_2	,,	,,	sq. in.
ρ_1	,,	,,	slugs per cu. ft. (1 slug $= 32{\cdot}2$ lb.).

The curve in fig. 77 shows the value of the coefficient C for a series

of different values of p_1 as observed in experiments upon a Zenith aero-engine carburettor of the form shown in fig. 78. Another carburettor of very different choke form and jet arrangement showed a slightly lower, and less constant, series of values for C, but one which lay between the limits 0·81 and 0·83 over the working range. The values of the coefficient given in fig. 77 include observations with and without petrol flowing in the jets, and at values of P equal to 14·5, 10·6, and 6·75 lb. per sq. in., and it may be accepted that these changes are without noticeable effect upon the coefficient. One

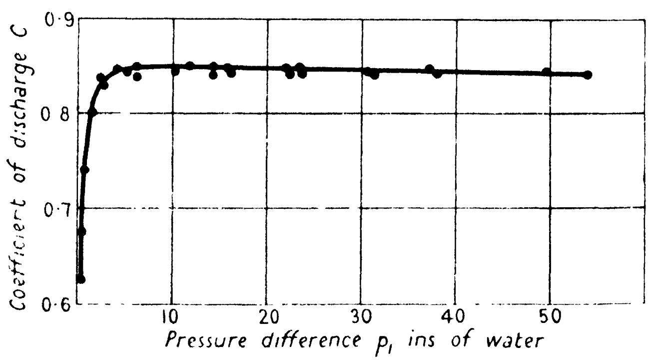

FIG. 77. Coefficient of discharge for air-flow in a typical carburettor, with and without petrol flowing, and with inlet pressures 14·5, 10·6, and 6·75 lb. per sq. in.

may conclude that in all practical carburettors having reasonably well-formed passages of the general type shown in fig. 78, the value of the coefficient will lie between 0·82 and 0·85, and that it will remain constant to within about 1 per cent. over the working range of values of p_1. In this connexion it is convenient to remember that a value of $p_1 = 5$ in. of water corresponds at ground-level to an air speed at the throat of very nearly 150 ft. per sec., and 20 in., therefore, to close upon 300 ft. per sec.

Another characteristic of equal interest in regard to the air-flow through a carburettor is the degree to which the inlet pressure is recovered after the air-stream has left the Venturi on the engine side. In the perfect Venturi giving pure streamline flow and no skin friction, the pressure recovery would be complete. The observed drop in pressure, which may amount to 3 or 4 per cent., is a measure of the loss through skin friction, and fluid friction in the air-stream arising from turbulence. Such a loss will obviously be much increased by the presence of jets and their supports, and by the discharge of fuel into the air-stream, all of which will tend to break up the streamline flow.

The curves of fig. 79 show how the degree of pressure-recovery depended upon the mass-flow, at three different values of the initial pressure P, in the same Zenith carburettor as before. It should be noted that the three curves A, B, and C are plotted to a common base of mass-flow in lb. per sec., and the appearance of a more rapid fall off in the pressure-recovery at the lower pressures is due to the reduced density of the air flowing. When compared at the same speed of flow the degree of pressure-recovery was substantially the same at the various pressures.

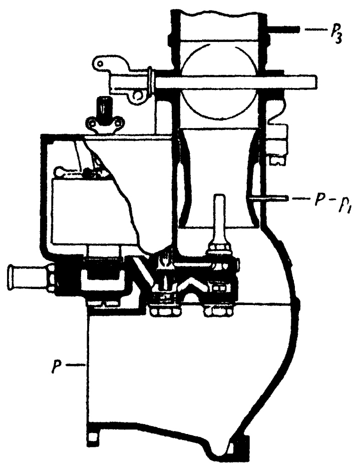

FIG. 78. Cross-section of an early type of Zenith aero-engine carburettor.

A high coefficient of discharge does not necessarily mean a high degree of pressure-recovery. Indeed, the two are largely independent. The coefficient of discharge is affected only by conditions before the throat of the Venturi. It represents the difference between the actual, and a theoretical frictionless, flow for a certain pressure-drop p_1 between the entry and throat, and may be said to be unaware of what goes on beyond that point. Pressure-recovery, on the other hand, although affected slightly by skin friction and turbulence before the throat, depends chiefly upon the turbulence in the expanding part of the air passage. Since the petrol is introduced into the air-stream *at* the throat, it will be clear why the discharge coefficient was unaffected by its presence, whereas the degree of pressure-recovery was markedly reduced when petrol was flowing, as illustrated by the dotted curves D, E, F in fig. 79 when compared with A, B, and C. The reason was that the introduction of the petrol increased the turbulence in the air-stream only beyond the throat. For the same reason it is found that the exact method of introducing the fuel into the air-stream, i.e. the design of the jet or 'diffuser', has a marked effect upon the degree of pressure-recovery.

The reason why conditions before the throat have only a slight effect upon the pressure-recovery, as compared with those which follow it, is that flow in a converging channel is essentially stable and tends to maintain a streamline character. Turbulence is slight, and the stability of the streamlines is in general unaffected by an increase of velocity, a fact which accounts for the constancy of the coefficient of discharge in fig. 77 over a wide range of values of p_1. In the diver-

gent part of the channel, on the other hand, the flow tends to instability and turbulence is much more readily set up by any obstructions; and moreover an increase of velocity is invariably accompanied by an increase of turbulence.

The falling curves of fig. 79 reflect the increase of turbulence which accompanies an increase of velocity and mass-flow in the

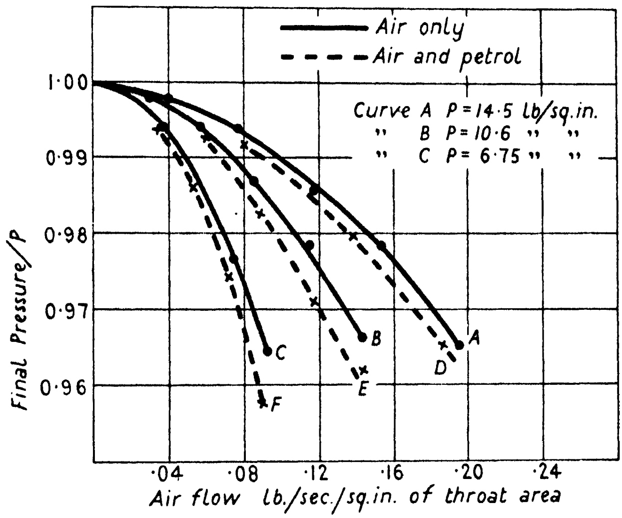

FIG. 79. Pressure-recovery at various rates of air-flow, with and without petrol flowing, and at three values of the inlet pressure P. Zenith carburettor of fig. 72.

divergent part of the air passage, as compared with the almost perfect constancy of the coefficient of discharge which is indicative of conditions in the earlier, convergent part.

ART. 53. *The effect of pulsating flow.*

There are two conditions involving unsteady flow in which the widely different densities of air and petrol will upset the mixture ratio established during steady conditions. The first, due to a sudden opening of the engine throttle, has already been mentioned. Here the necessity of maintaining a normal, or somewhat rich, mixture for rapid acceleration is so imperative that it justifies the inclusion of a special device by which a limited supply of extra petrol is injected, sufficient for the few seconds which follow the sudden opening of the throttle.

The need for the extra petrol is always associated with a definite throttle movement, and therefore the most suitable device is some

positive form of displacement pump, which overcomes the effect of petrol inertia, operated by a movement of the engine throttle itself. The working of the device will be clearly understood from the description of an actual modern carburettor to be given in art. 55. The details may vary in different designs but the principle is always that of providing, in some positive manner, an extra supply of petrol just while the throttle is being opened.

The other departure from the condition of steady flow assumed in the last article arises from the intermittent pumping action of the several cylinders drawing air from one carburettor. There are two sides of the question to be considered: the effect of the pulsating flow upon the quantity of air passed by the carburettor, and its effect upon the proportionate flow of petrol and air. Both effects will be less and less important the greater the number of cylinders supplied from one carburettor and the greater the revolution speed. It is well known, of course, that in a single-cylinder engine quite a substantial degree of supercharging, up to about 2 lb. per sq. in., can be obtained by fitting a long induction pipe, in which the natural period of vibration of the air column synchronizes with the r.p.m. of the engine. The greatest effect is produced when the length of pipe is half the natural wavelength of the air-pulsations, so that a node is formed at the pipe entrance and an antinode (point of maximum pressure variation) at the carburettor.

In a multicylinder engine there are never likely to be less than three cylinders drawing air through a single Venturi or 'choke' tube, and moreover there is of necessity a considerable volume in the induction system between the choke tube and the cylinders. This fact will much diminish any pressure fluctuations at the throat, and normally the intake pipes through which air is supplied to the carburettor are not of a length to set up any appreciable pressure effects of resonance. Forced vibrations may, however, be set up within the manifold, between the carburettor and the cylinders, and these might not only affect the total air-flow through the carburettor, but the distribution of the air between the cylinders. Both effects would be eliminated if the point of attachment of the carburettor to the manifold could be arranged to be a node of the forced vibrations set up, but manifold design has usually to be based upon other considerations, and it remains to examine experimentally what are the likely magnitudes of the effects of pulsating flow in manifolds as they are.

The information available is somewhat scanty, for although the experiments[44] covered a wide range of periodicities in the pulsation, and three different amplitudes of pressure fluctuation, only a single

'manifold volume' in the form of a straight pipe 15 in. long was employed. Of the three amplitudes, the minimum is stated to have corresponded reasonably closely to the condition of a single carburettor supplying eight or more cylinders, while the maximum amplitude approached that which would occur when a single car-

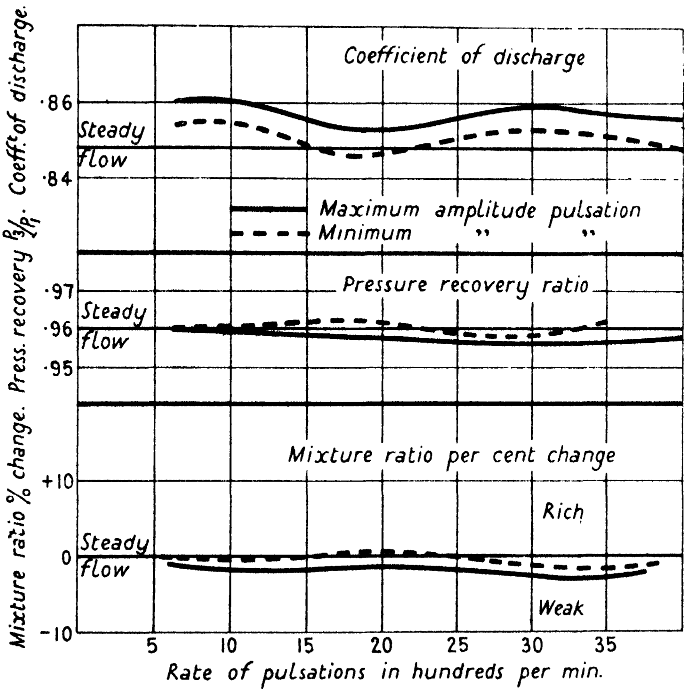

FIG. 80. Effect of pulsating air-flow in the Zenith carburettor shown in fig. 78.

burettor is made to supply only three cylinders, of which the pumping strokes occur at regular intervals.

The effect of the pulsations upon the coefficient of air discharge, upon the pressure-recovery ratio, and upon the mixture ratio, are summarized in fig. 80 for the Zenith carburettor of fig. 78, over a range of periodicities from 800 to 4,000 pulsations per min. All the observations were made at the same value of the mass-flow of air, and the corresponding results for steady flow conditions are given for comparison.

It will be noticed, in the first place, that the effect of the pulsations was slightly to increase the coefficient of discharge at all periodicities as compared with the value for steady flow, while showing also a roughly harmonic variation with the periodicity. The greatest

increase of the coefficient was just over 1 per cent. The mixture strength shows a variation which moves with that of the discharge coefficient but is rather greater, amounting to a weakening of the mixture of between 2 and 3 per cent. over the range from 2,700 to 4,000 pulsations per min.

With three cylinders drawing from one carburettor this would mean a range of engine speeds from 1,800 to 2,600 r.p.m., so that these results may probably be accepted as a fair guide to what may be expected in practice under the worst conditions, with this and similar types of carburettor.

With any radically different design the results might differ substantially, more especially as regards the effect of pulsations upon the mixture ratio. For example, in the Zenith carburettor it is only changes in the air pressure at the throat which affect the fuel-flow. The fuel metering is only slightly disturbed and, such as it is, the alteration of the ratio is consistent with the change in the apparent throat coefficient. In a second carburettor tested at the same time, however, the air space in the float chamber was in communication with the pressure on the exit side of the Venturi. The petrol flow was therefore affected, not only by the discharge coefficient, but also by the degree of pressure-recovery, and was much more sensitive to the periodicity of the pulsations. Between values of 2,300 and 3,500 per min. the mixture changed, in this other carburettor, from 4 per cent. weak to 3 per cent. rich, although the mean value of the mixture ratio over the whole range of periodicities was, unlike the Zenith carburettor, the same as for steady flow.

In conclusion it should be mentioned that any closure of the throttle, assumed to be on the engine side of the Venturi, rapidly reduced the amplitude of the pressure fluctuations to a negligible amount. In supercharged engines, moreover, when the supercharger is between the carburettor and the engine, the same will be true, for not only will the supercharger serve to protect the carburettor from pressure pulsations, but also in such engines the whole number of cylinders are in effect served from one carburettor, via the supercharger, so that in any case the amplitude of any pressure fluctuations would be very small.

While, therefore, in some circumstances, pulsating air-flow may have a quite noticeable effect upon the fuel-metering characteristics of a carburettor, this is only likely to be so in a normally aspirated engine at or near full throttle. Under these conditions, as we have seen, the mixture would normally be 10–20 per cent. rich in any case, so that a variation of 3–5 per cent. is not likely to affect the engine's behaviour.

ART. 54. *Adjustment for altitude.*

Although it might be possible, apart from certain practical difficulties, for the simple carburettor of fig. 73 to maintain a constant fuel-air ratio so long as there was no change of air density, it is clear from equation (42) that any increase of height must lead to an increase in μ, approximately in inverse proportion to the square root of the air density.

There must always remain, therefore, the need to provide some form of compensation to counteract this tendency to enrichment. Some degree of 'load compensation' is in practice also needed to allow for differences of petrol level and for the viscosity effects of the liquid at a metering orifice with chamfered ends. How this load compensation is contrived in a practical carburettor will be described in art. 55. In the present article it will be supposed that these effects of level and viscosity have been provided for, and that the carburettor is capable of giving a constant fuel-air mixture at all throttle openings, apart from the enrichment under certain conditions which is provided by special devices.

The question of adjustment for altitude must be considered from three quite separate points of view:

(1) How is the fuel-air ratio which the engine requires affected by different conditions of external pressure and temperature?
(2) How is the normal functioning of the carburettor affected?
(3) How can the carburettor be made to meet the wider range of working conditions?

Taking first the question of engine requirements, explosion experiments in closed vessels[45] provide evidence that with fuel-air mixtures which are not much weaker than the correct one, the effect of a lowering of the initial pressure upon the rate of burning of the charge is not great, and, moreover, that the ratio of increase of the pressure on explosion for a given mixture is very nearly a constant, irrespective of the initial pressure. These were experiments with a stagnant mixture, of course, but the effect of the turbulence in an engine would be rather to even up the rates of burning than to do the reverse. As regards the fall of the atmospheric temperature with altitude, although this amounts to about 2° C. per 1,000 ft. the lowering of the gas temperature in the cylinders at the moment of ignition would be much less, and too small to have any effect on the rate of chemical reaction.

There is, therefore, no reason to anticipate on chemical grounds that the fuel-air ratios required at high altitudes will be any different from those at ground-level, and all engine tests carried out under

altitude conditions go to support this conclusion. Experiments at the Bureau of Standards[46] covered ranges of pressure corresponding to heights up to 30,000 ft., as well as tests at different speeds, loads, and compression ratios. The conclusion reached was that the optimum air-fuel ratios, both for maximum power and for economical operation, were unaltered by a change of height. The majority of these tests were made with a constant air temperature of $+10^\circ$ C., so that they were tests of the effect of a change of pressure rather than of altitude. Another series of tests at a constant pressure corresponding to 5,000 ft., and for ranges of temperature from -20° to $+40^\circ$ C., showed that the maximum power was obtained at an average air-fuel ratio of 13·5 : 1 and maximum economy at 18 : 1 throughout all the temperatures tested. The economical mixture was taken as that which gave a drop of 5 per cent. below the maximum B.M.E.P. This would correspond to a drop of about 2½ per cent. in r.p.m. when flying level, without any change of throttle position.

More recent experiments in this country,[47] to be discussed more fully in Chapter XIV, were done under full altitude conditions as to both pressure and temperature up to 23,000 ft. There was in these experiments some evidence that the proper evaporation of the petrol, and good distribution between the cylinders, was sufficiently affected by the low temperatures to necessitate a rather lower *average* air-fuel ratio above 10,000 ft. This necessary enrichment at altitude, however, is due, not to any essential influence of pressure or temperature on combustion, but purely to the failure to secure a uniform mixture of the fuel and air after leaving the carburettor, on account of the sluggishness of the evaporation, which is probably far from complete even towards the end of the suction stroke.

Evaporation of the fuel at high altitudes is affected by the pressure as well as by the temperature of the air. In a mixture containing a constant ratio by weight of air to fuel vapour the ratio of the partial pressures* of the air and fuel vapour will be constant whatever the total pressure. The partial pressure of the fuel will therefore be lower during evaporation at high altitudes, and the effect of the lower temperature is to some extent compensated by the lower pressure, which assists evaporation.

In fig. 81 the dotted curve shows the relation between the pressure and temperature in the standard atmosphere, with the corresponding

* The partial pressure of a fuel in a certain proportion by weight with air, when the fuel is a mixture of hydrocarbons of very varying molecular weight, will depend upon a mean volatility. In Wilson and Barnard's work, referred to below, a series of curves are given in which the partial pressures of fuels in different mixture ratios with air are related to a certain average boiling-point.

heights indicated. The full-line curves indicate, by the temperature scale, at what points a full-power mixture and an economical mixture of a light aviation petrol would begin to deposit liquid at each value of the total pressure.[48] The distillation curve of the petrol is given above. The 12 : 1 mixture, for example, would begin to condense at $-4°$ C. at atmospheric pressure, whereas it would have to be cooled to $-17°$ C. before it began to do so at 15,000 ft. On the

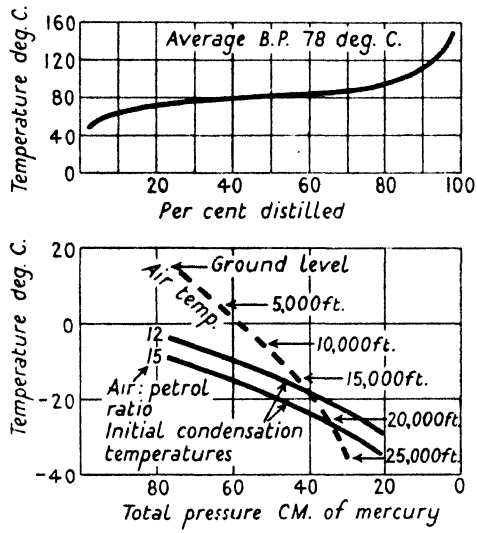

FIG. 81. Condensation temperatures at different altitudes for a full-power and an economical mixture of air with aviation petrol.

other hand, the normal atmospheric temperature is 19° C. above the condensation point at ground-level, but only 2° C. above it at 15,000 ft. and 12 ° below it at 25,000 ft. It is very clear, therefore, that the conditions are far less favourable for complete evaporation in the manifold at great heights, in spite of the assistance derived from a lower total pressure.

Turning now to the question how the carburettor can be made to maintain a constant mixture ratio under widely different conditions of air pressure and temperature, one can aim either at some form of manual control which must be adjusted at each height, using the engine as a rather crude indicator of the correct adjustment for maximum power or the economical mixture; or one can go for the completely automatic carburettor in which the change of external air pressure is itself the effective agent for adjusting the carburettor to function in a manner appropriate to each height, so that a constant mixture ratio is maintained.

What the necessary adjustments to the carburettor amount to, and how they can be effected, must now be considered. Referring to equation (42), and neglecting the effect of air compressibility which has been shown to be small, then the mixture ratio, μ, can be expressed as

$$\mu = (\text{constant}) \times \frac{a}{A_2}\left(\frac{p_2 \sigma}{p_1 \rho_1}\right)^{\frac{1}{2}},$$

in which the constant includes the coefficients of discharge of the air Venturi and the petrol-metering orifice. By means of designs which compensate for the viscosity effects in the fuel, to be described in the next article in connexion with the practical carburettor, it is possible to maintain these coefficients of discharge very nearly constant over the working range of p_1 and p_2. There remains, however, the possible influence upon the value of the discharge coefficient of a change of temperature of the fuel, and therefore of its viscosity.

By comparison of figs. 75 and 76, and allowing for the diminished effect in an orifice of a more favourable L/D ratio than that of fig. 76, a reduction of 4 per cent. in the petrol flow for a 20° C. drop in the temperature appears to be an outside figure. It is impossible to say what reduction of petrol temperature is liable to be experienced, for it depends upon the arrangement of the storage tanks and the duration of the flight, but considering the nearness of the carburettors to the engine it seems unlikely that temperatures more than 20° C. below the ground-level starting temperature will often be reached. In any case, such a fall in temperature is to be associated with an increase of height and may properly be considered in connexion with the compensation for altitude, towards which the reduction of petrol flow due to lowered temperature forms a natural contribution.

The rest of the altitude compensation, of which the aim is to maintain μ constant for all values of the air density, ρ_1, can only be attained by suitable adjustment of one or both of the ratios a/A_2 and p_2/p_1, the one being known as the 'variable jet', and the other as the 'pressure balance' system. Both methods have been used for manual mixture control, and if one is prepared to put up with adjustment of the mixture control every time, after the throttle has been set for cruising under steady conditions, then there is not much to choose between the two systems.

From the point of view of convenience, however, and still more if we have in view the possibility of making the altitude compensation automatic, there is an important difference between the two methods. Automatic mixture control must rely upon some device

which sets the control in a definite position, determined by the altitude; for example, by means of an aneroid box which moves in a definite relation to the atmospheric pressure. For manual control, also, there is clearly an advantage in being able to graduate the setting of the control in terms of height, so that it can be set to a definite correct position after simply reading the altimeter.

For this to be possible, however, the action of the control must be unaffected by the position of the throttle, so that a fixed percentage reduction of the fuel supply is achieved, according to the height, and the same at all throttle positions. If the compensation is achieved by adjustment of a/A_2, then it will not be affected by the throttle provided only that the flow characteristics of the petrol orifice (i.e. the variations of its discharge coefficient) are not affected by a change of its area a. This is not altogether an easy thing to accomplish, but it is very much easier than to arrange a pressure balance system of control which would be equally independent of the throttle position. The latter, indeed, would be a virtually impossible thing to arrange. If any slight reduction of p_2 necessary for altitude compensation is achieved by sealing the carburettor float chamber and connecting its air space to some point near one end of the air Venturi, where the pressure will be slightly sub-atmospheric, then this must necessarily be a fixed point, and the pressure at a fixed point along the Venturi axis bears no fixed relation to the pressure at the throat, so that the relationship of p_2 to p_1 would vary with the mass-flow in an uncontrolled manner. Or again, if the value of p_2 is settled by a balance maintained between a small adjustable connexion between the float chamber and the Venturi throat, and a small 'leak' of air into the float chamber from outside, then this balance will equally be upset by a change in the value of p_1 due to a change of throttle.

It comes to this, therefore, that altitude compensation by the variable jet method is essential for the working of an automatic control, and has the great advantage where a manual control is employed that the mixture control lever can be arranged to have a definite, correct, position for every altitude.

ART. 55. *The carburettor in practice.*

It is proposed in the present article to examine how the principles we have been discussing can be embodied in the design of a modern carburettor, and finally to give an illustration of an actual design. There have been some designs which could scarcely claim to have been influenced by rational principles, but no such remark would apply to the Stromberg carburettor illustrated in figs. 87 and 88. It is an example of a very compact and practical piece of equipment,

designed, at each point, so far as circumstances allow, in accordance with scientific principles.

The design of a modern carburettor is highly complex, and in order to see the reason at the back of its complexity it will be well to recapitulate all the separate aims which have to be met.

(1) There is, first, the *main fuel-metering system*, whose business it is to provide an unvarying fuel-air mixture at all throttle openings.
(2) *The idling system*, designed to supply a slightly rich mixture when the throttle is almost closed and the engine is 'ticking over'. The in-going charge is at a low pressure under these conditions and is also much diluted with residual exhaust gas. In order to get satisfactory combustion, the ratio of the partial pressures of the fuel and air must be that at which the speed of burning is a maximum.
(3) *The accelerating system*, for providing a rich mixture just when the throttle is opened. As explained earlier, some positive method of supply is favoured, in order to eliminate the effect of the inertia of the petrol and the tendency to a weak mixture when the throttle is opened suddenly.
(4) The system for providing full-throttle enrichment, also called the *economizer system* because it allows an economical mixture to be maintained except while it is in operation, at and near full throttle.
(5) *The mixture control system* for altitude compensation.

In a carburettor with special arrangements for full-throttle enrichment it is the business of the main metering system to maintain a constant fuel-air ratio at all values of the mass-flow of air within the working range, for a given air density. The working range in an aero-engine would be from full throttle, full speed, conditions, down to about 30 per cent. of the mass-flow of air under those conditions. This would correspond to throttled operation at the minimum cruising speed.

Having accepted a type of fuel orifice in which viscosity effects are not absent, some form of load compensation will be necessary; and, since the uncompensated jet shows a discharge coefficient which decreases with the mass-flow (see fig. 75), compensation will take the form of providing some increase of the relative fuel-flow above that provided by the simple jet when the throttle is partially closed.

In the well-known Zenith carburettor a second crifice is provided, working in parallel with the main jet, the flow through which is scarcely influenced by the depression at the Venturi and the air-flow.

Under full-throttle conditions the petrol supply is nearly all through the main jet, but when the air-flow and the depression at the Venturi diminish as the throttle is closed, the nearly constant flow through the compensating orifice becomes an increasing fraction of the whole, and makes up for the drop in the discharge coefficient controlling the flow through the main jet.

While this double jet system gives satisfactory load compensation, it suffers from the drawback of a main metering orifice with a large L/D ratio and with a discharge coefficient, therefore, very much subject to temperature variations in the petrol. A more satis-

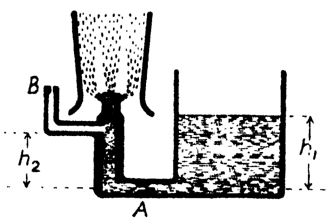

FIG. 82. Submerged metering orifice and main jet with 'air-bleed' compensation.

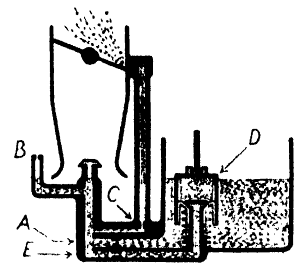

FIG. 83. Slow-running jet and pump to provide for sudden acceleration.

factory method, because less subject to the effects of temperature, is that employed in the Stromberg carburettor, where compensation is achieved within a single jet by making use of the 'air-bleed' principle.

This will be understood from fig. 82. The main metering orifice is submerged, at *A*. A pipe introducing the air-bleed joins the main jet about half-way up, and air which is sucked in through the orifice at *B* passes up to the Venturi throat as a series of small bubbles, thus emulsifying the petrol and assisting atomization in the main air-stream.

If p' represents the pressure difference across the orifice at *B*, and p_2, as before, the difference across the main fuel orifice at *A*, then

$$p_2 = p' + (h_1 - h_2).$$

p' will be very slightly less than p_1, by the effective head of petrol in the main jet above the air-bleed. This would be difficult to estimate, but is certainly very small, and we may take p' as equal to p_1 with sufficient accuracy. The petrol flow, instead of being proportional to $(p_1)^{\frac{1}{2}}$ will now be proportional to $(p_1+c)^{\frac{1}{2}}$, where c is a constant. In other words, it will diminish less rapidly than the simple jet for a certain change of p_1, and by adjustment of the design the air-bleed jet

can be made to pass a mass-flow of petrol very nearly in constant ratio to the mass-flow of air at all values of p_1.

The size of the main jet passage above the air-bleed is also a factor in the fuel-flow, for by reducing the bore of this passage a restriction upon the flow of the air-petrol emulsion can be introduced as the throttle is opened, and this will also serve to counteract the increasing discharge coefficient of the main metering orifice.

Turning now to the idling system, fig. 83 shows this diagrammatically, in action, with the throttle almost closed. There is no suction in the main jet, and the petrol level both there and in the air-

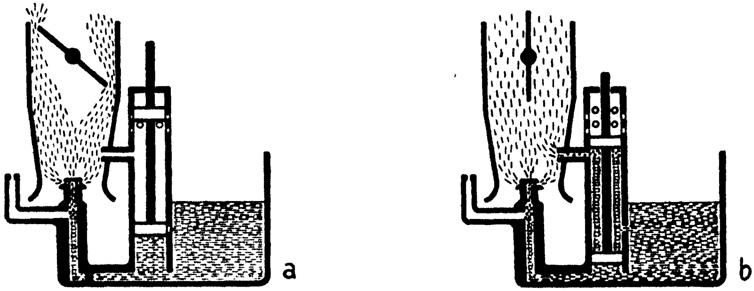

FIG. 84. Method of providing enrichment of the fuel-air ratio at full throttle by a double piston operated by the throttle. (*a*) Cruising position. (*b*) Full-throttle position.

bleed connexion has risen to that of the float chamber. The idling jet is subject to the high vacuum above the closed throttle, and the correct quantity of fuel is metered at the orifice *C*.

The acceleration system is illustrated in the same figure. The sleeve *D* surrounds a movable piston which is held up against the under side of a fixed mushroom valve by a spring. The vertical movement of the sleeve *D* is controlled by the throttle lever, so that when the throttle is opened the sleeve is pressed down, forcing the piston down with it and expelling petrol through the orifice *E* until the piston is restored to position by the spring and the valve is again closed. By the combined action of the sleeve-movement and the spring, an extra supply of petrol is provided during a second or two following an opening of the throttle. The extra supply is forced through the orifice *E* under the diminishing pressure produced by the spring, and the more rapid the throttle opening the greater will be the initial rate of supply; which is just what is required to compensate for the inertia of the petrol in the main supply system.

Two methods of effecting the full-throttle enrichment are illustrated in figs. 84 and 85. In fig. 85 a needle-valve is lifted from its seat by the final stage of movement of the throttle lever; and in

fig. 84 the same movement lowers the double piston from the position shown at (*a*) to that at (*b*), thereby uncovering an extra orifice which admits petrol to the space between the two pistons. Air is admitted at the same time through holes in the upper piston, to form an emulsion with the petrol before it is drawn into the main air-stream.

The various additions to the simple carburettor illustrated in figs. 83–5 have been for the purpose of producing enrichment in special circumstances and have in no way affected the functioning of the

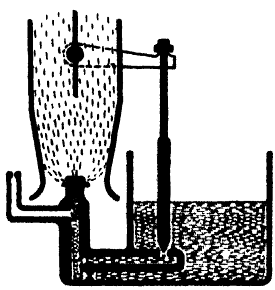

FIG. 85. Method of providing enrichment at full throttle by bringing in a second orifice in parallel with the main jet.

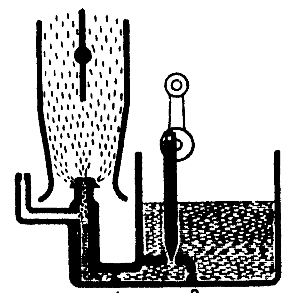

FIG. 86. Variable jet type of mixture control for altitude compensation.

main jet, with its metering orifice, shown at *A* in fig. 82. When it comes to compensation for altitude, however, it is the main metering system to which this compensation must be applied. The variable jet type of mixture control is illustrated in fig. 86. Under ground-level conditions with control in the full-rich position the needle is lifted away from its seat and the petrol is metered at the orifice *A*. As the needle is gradually lowered to compensate for altitude, resistance is introduced into the petrol-flow circuit until finally the supply to the main jet is controlled by the two orifices *A* and *B* in series. In a control of this kind it is possible to keep the characteristic change of the discharge coefficient of the system the same, while varying the effective area of orifice.

Finally, in figs. 87 and 88 it can be seen how the features which have been illustrated diagrammatically have been worked into an actual carburettor. The float chamber is not shown. It exhibits no special features, except that it is ingeniously designed so that the depth, below the free surface, of the passage which conveys the fuel to the metering system is always the same, whatever may be the angle in pitch of the aeroplane, whether climbing or diving.

It will be noticed that every metering orifice is placed in a submerged position, and that the proportions of each are about the same, the L/D ratio being in the neighbourhood of unity. Both of these points, as we have seen, are factors which assist in reducing the irregularities of metering to a minimum.

ART. 56. *The direct injection of fuel.*

However perfectly a carburettor may be able to control the relation between the total amounts of fuel and air which enter the manifold of an engine, it cannot ensure the same perfect proportion in each cylinder. Some tests to be referred to in Chapter XIII have shown that in a well-designed manifold the equality of the distribution at ground-level is surprisingly good; but with low atmospheric temperatures and sluggish vaporization of the petrol there must be times when the distribution breaks down badly, and over this the carburettor has no control at all.

It is therefore natural that the development of the high-speed Diesel engine, and of pumps capable of delivering small and accurately controlled quantities of fuel to each cylinder, every cycle, should have led designers to contemplate their employment in place of a carburettor serving a number of cylinders. There is, too, a more potent reason in favour of the direct injection of petrol. In every carburettor the air and fuel passages may, when the humidity and atmospheric temperature are favourable, become blocked with ice. Many mysterious engine failures may undoubtedly be attributed to this cause,* and among air-mail carriers which fly in all weathers it is a very real menace. Direct injection of the fuel would be made during the suction stroke, either into the cylinder itself or into the manifold close to each inlet valve, and all trouble from ice formation would be completely eliminated.

It has been argued, also, that the absence of a carburettor choke-tube, and of all need for heating the ingoing air to facilitate fuel evaporation, would lead to a higher volumetric efficiency and to more power. Experiments have shown, however, that in aero-engines the probable gain on this account is not important, although it may well be more so in road-vehicle engines, in which the air-velocity through the carburettor choke is normally very great at full throttle and a high speed.

Apart from the elimination of the freezing danger, which may possibly be controllable in other ways, there seems little to justify the added complexity and expense of a multiplicity of small pumps

* By the time the engine can be examined all trace of the cause of failure will usually have disappeared.

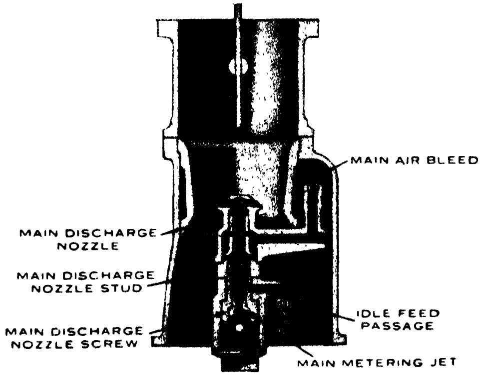

FIG. 87. Section of Stromberg Carburettor, showing main jet and air-bleed. Slow-running jet not shown.

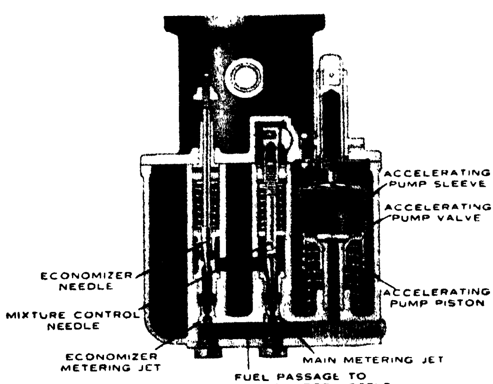

FIG. 88. Section of Stromberg Carburettor, showing accelerating pump; mixture control for altitude; and economizer needle, i.e. enrichment for full throttle only.

for dealing with a volatile fuel, when the stationary carburettor can be made so satisfactory. Nevertheless an interesting report[49] has recently been published upon single-cylinder research work using injected fuel of a less volatile type than is demanded by the carburettor, combined with a special valve timing in the engine.

The fuel was of a narrow boiling range, between 155° and 205° C., and of a type produced by hydrogenation of the crude oil. The anti-knock value was very high, and it was possible to use compression ratios up to 7 : 1 without detonation; and even up to 9 : 1 in certain conditions. It is claimed that the danger from fire with a fuel of this volatility would be much reduced, and while it is open to doubt whether the ignition of this fuel would be any less likely than petrol in the event of a crash, its lower volatility should certainly prevent the fatally instantaneous spread of a petrol fire.

The object of the special valve timing was to provide a large valve overlap and so to obtain increased power by scavenging the combustion space and replacing the residual burnt products by air. As compared with a normal valve overlap of about 30° the special valve timing was as follows:

Inlet opens 70° before top centre. Exhaust opens 55° before bottom centre.

Inlet closes 45° after bottom centre. Exhaust closes 60° after top centre,

thus giving an interval of 130° during which both inlet and exhaust valves were open together. There is no waste of fuel with such an arrangement, since the fuel is injected after the closure of the exhaust valve, but its use in conjunction with a supercharger would mean some increase of the power for driving the latter, to provide any given supercharge pressure. The wastage of air would be more serious at high altitudes, for if the valve overlap were large enough to give the desired scavenge at sea-level with, say, a boost pressure of 2 lb. per sq. in. above atmosphere, then there must be a wastage of air when the pressure difference across the valves becomes 6½ lb. per sq. in. as it would do at 10,000 ft.

From the results of these tests, which are illustrated in fig. 89, it will be seen that even when operating with atmospheric pressure on the inlet side there was a gain of 10 lb. per sq. in. in the B.M.E.P. caused simply by the increase of the valve overlap from 30° to 130°. This can only mean that a resonance effect in the inlet and exhaust pipes was providing some degree of scavenging even without a positive boost pressure on the inlet side.

The possibility of combining scavenging with a high compression

ratio and a substantial boost pressure, without trouble from detonation, certainly gives outstanding results in the way of power output as shown by the curve of B.M.E.P. in fig. 89. With a boost pressure of about 3 lb. per sq. in. the output of the cylinder even at 1,750

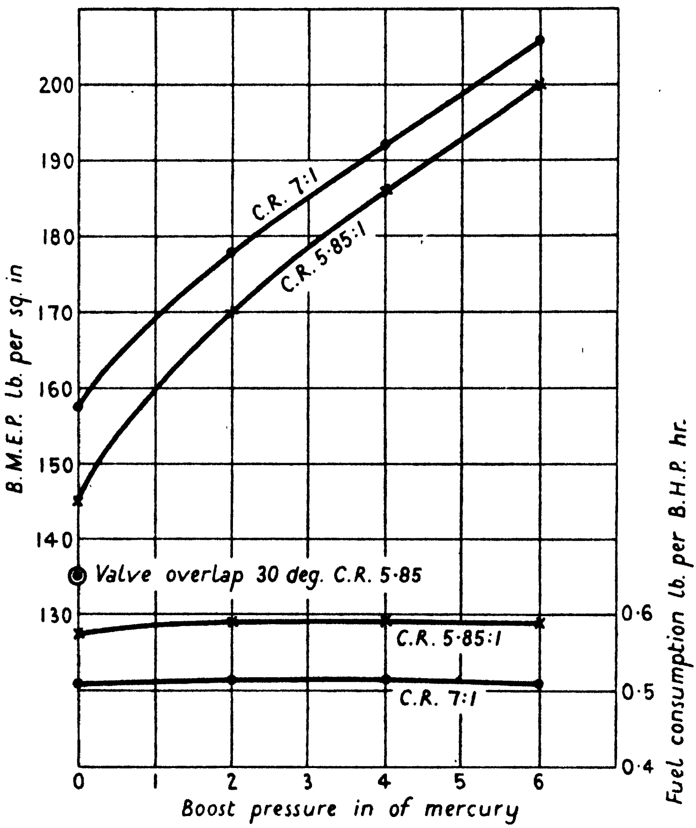

FIG. 89. Power and fuel consumption obtained with direct injection and hydrogenated fuel. 1,750 r.p.m.; 130 deg. valve overlap; ignition advance 30 deg.

r.p.m. was nearly 28 B.H.P. per litre. The fuel consumption suffers by comparison with tests using a volatile spirit supplied through a carburettor. At 7 : 1 compression the minimum fuel consumption per I.H.P. hour ought to be 0·37 lb. (see fig. 137, p. 363) and the corresponding brake figure, therefore, 0·41 at a mechanical efficiency of 90 per cent. At such very high B.M.E.P.s the mechanical efficiency of the engine can hardly have been less than 90 per cent., and even an allowance of 20 per cent. increase in fuel for the full-power mixture strength would only bring the specific consumption up to

0·49 lb. per B.H.P. hour. The observed specific fuel consumption of 0·51 lb. must be put down to unequal distribution of the fuel throughout the compression space in the cylinder, which results in its incomplete combustion.

Herein lies the weakness of the whole project of burning injected fuel of low volatility in a high-speed engine; a weakness which affects not only the efficiency with which the fuel can be burnt, but also the problems of hardly less importance in the aero-engine, of ensuring easy starting and steady 'idling', when throttled right back.

In the experiments described above those difficulties were reduced to a minimum through the homogeneity of the fuel employed. A boiling range between 155° and 205° C. is a very narrow 'cut', such as would usually mean an increased cost of production. If the high flash point of the fuel is to be preserved, to reduce the fire risk, then the more volatile products must be rigidly excluded, and any extension of the boiling range in an upward direction to assist production would very rapidly increase the difficulties of securing proper combustion to an indefinite degree. It is clear from such reference as there is in the report to trials with alternative fuels, that the volatility and flash point were critically important.

X

SUPERCHARGERS

Art. 57. *Types of supercharger.*

The air compressors which have been employed for the supercharging of internal combustion engines fall into three classes which differ widely in their principles of operation. They are:

(1) The vane type.
(2) The simple displacement type.
(3) The centrifugal type (*a*) gear driven; (*b*) exhaust-turbine driven.

The operating principle of each type is illustrated diagrammatically in figs. 90 to 92. The vane type is a form of displacement compressor, but differs from the simple Root's machine in that the air undergoes a certain degree of compression within the casing between intake and delivery. Fig. 90 shows in outline the mechanism of the Powerplus supercharger, a design which has been employed with success on racing motor-cars, and of which experimental models have been built for aero-engines. The air enters at *A* and is trapped between the casing and two successive vanes. As the drum *B* revolves, the air is compressed because the space between the drum and the case *C* diminishes. The compressed air is discharged as soon as a vane-tip passes the lip *D* on the delivery side. As the drum *B* revolves it exerts a side pressure on the vanes, which are mounted on eccentrics rotating round a fixed centre, but a different one from that of the drum *B*. The vanes can slide in the narrow slots in the drum surface through which they pass. The movement of the vanes in relation to the fixed casing is controlled by their rotation on the two eccentrics, and is such as to preserve a small clearance between the ends of the vanes and the inner surface of the case. The clearance must be as small as possible, to prevent leakage of air back to the low-pressure side of the blower.

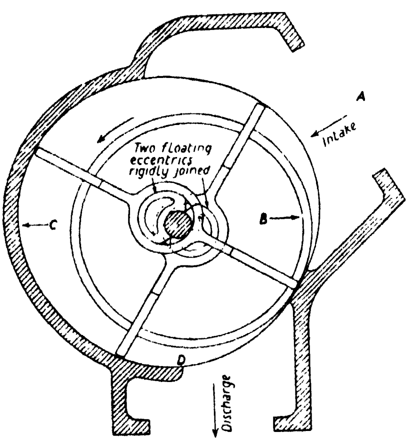

Fig. 90. Powerplus vane type of supercharger.

The advantages of the vane type of compressor over the Root's design are theoretical rather than practical. As mentioned above, there is some compression of the air within the blower casing, and this enables the vane type to show a higher compressing efficiency when the pressure ratio is higher than about 1·4. On the other hand, for a given size of compressor the quantity of air passed at a given speed is less than by the Root's design, and in reliability it is unlikely that the vane type could ever compete with the surpassing simplicity of the Root's.

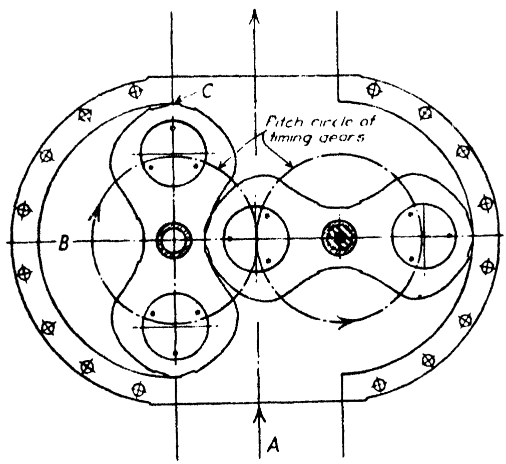

FIG. 91. Root's supercharger.

The latter, illustrated diagrammatically in cross-section in fig. 91, consists essentially of two symmetrical displacing elements which rotate within a casing in opposite directions, while geared together externally. They never touch one another, nor the casing, but are designed, like the vane type, with very fine clearances to avoid leakage of the air back from the high- to the low-pressure side. As designed for an aero-engine, both the case and the impellers are made of an aluminium alloy, and this similarity of material facilitates the maintenance of the proper clearances when the supercharger heats up in service.

The vane type of supercharger suffers from the changes of acceleration undergone by the moving parts, and the forces needed to produce them at high speeds. In the plain rotation of the independent elements of the Root's compressor unbalanced inertia forces are absent, and it has been possible to operate the type up to 7,000 r.p.m. as compared with a maximum of about 2,500 for the vane type. The drawback of the Root's is that after the air has

entered at *A* and been trapped in the space *B*, it remains at the intake pressure until the tip *C* of the rotor passes the corner of the casing, when high-pressure air from the discharge side rushes back and compresses it in the space *B*. From that point delivery of the air is against the full discharge pressure, and in consequence the work done in compressing and delivering each lb. of air is high (see below). From fig. 91 it will be seen that there are four discharge pulsations per revolution, that is, two for each of the two impellers.

The centrifugal supercharger consists of a simple rotating impeller *A*, usually with radial vanes, enclosed within a casing containing the stationary 'diffuser' blades *BB* (fig. 92). The working speed range in aero-engine designs is very high, 15,000–25,000 r.p.m. The air enters near the impeller axis, and while flowing out between the radial vanes it is given a high tangential velocity by the time it leaves the impeller periphery. The kinetic energy so acquired is converted into pressure energy as the air flows in a spiral direction between the fixed diffuser vanes, with a rapid reduction of its peripheral velocity.

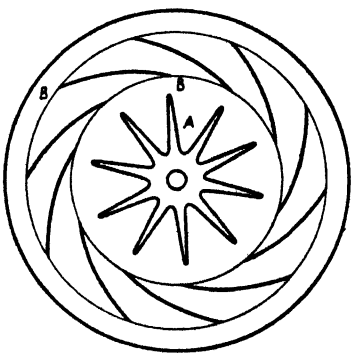

FIG. 92. The centrifugal supercharger.

When gear-driven from the engine the speed ratio is often about 10 : 1, and to avoid stripping the gears some form of slipping clutch has to be incorporated in the supercharger drive to provide for the huge inertia effect of the high-speed impeller when sudden changes of engine speed occur. When driven by an exhaust-gas turbine the impeller is coupled direct to the turbine wheel, which is of the simple Rateau type acted on by four or more jets of exhaust gas at a high velocity; the gas velocity having been acquired in flowing through nozzles of an appropriate shape from a receiver, usually at 2 or 3 lb. per sq. in. above normal atmospheric pressure.

Of the types of supercharger enumerated, the gear-driven centrifugal fan type is the only one which has reached the stage of extensive practical application in the air. The Root's and vane types, apart from the question of reliability, would be relatively heavy and bulky, and not nearly so readily embodied in the engine design as the fan type of compressor. The exhaust-turbine driven type is a most attractive system theoretically for the attainment of maximum engine power at great heights, for instead of absorbing useful crankshaft power in driving the supercharger, this is done by the energy left in

the exhaust gas, which is normally wasted; and, moreover, the reduction in the external atmospheric pressure, as heights increase, produces automatically a higher turbine speed, and so increases the compression ratio to compensate for the reduction of the pressure at the intake. The drawback to the scheme is that the additional 2 or 3 lb. per sq. in. of back pressure in the exhaust-gas receiver, before it passes to the turbine, imposes some extra heating of the engine exhaust-valves; and, moreover, the necessary pipe-work for conducting the exhaust gas to the turbine is a difficult constructional problem. When the design of the engine is such as to allow a neat and compact installation of pipe-work and turbine, then the heat concentration is liable to be such that the whole pipe system and turbine casing is maintained at a bright red heat. The mechanical and metallurgical problems of producing and maintaining the whole supercharger plant are most formidable, as also would be that of reducing the fire risk to a reasonable level.

It is for these and other reasons, such as the difficulty of avoiding the distortion of the turbine and supercharger casing caused by large temperature differences from point to point, that the exhaust-driven supercharger has not yet developed into a piece of practical equipment, in spite of the fact that an experimental design was applied to the well-known Napier Lion engine and successfully took a fully-loaded aeroplane to 32,000 ft. as long ago as 1926.

The gear-driven supercharger is capable of maintaining ground-level power to heights up to 12,000 ft. at a cost in extra weight of 40 or 50 lb. As employed in most installations the business of the supercharger is to maintain the pressure in the induction manifold constant at all heights up to a certain 'rated height' (see art. 77).

Under all conditions below this height it is necessary to throttle the air at its entry to the supercharger, and it is this method of control which is the chief drawback of the type. If constant engine power is maintained to the full supercharged height the compressor must provide roughly the same number of pounds of air per minute all the way up, and ideally should compress this in a gradually increasing ratio as the atmospheric pressure falls. The fan, however, is direct driven from the engine, and so long as it is maintained running at nearly constant speed by gearing to the crankshaft, the centrifugal supercharger must always be giving its full compression ratio. This means that although the engine receives very little benefit from the supercharger at low heights, the mass-flow of air through it, and the compression ratio, are constant, and the power absorbed in driving it is no less than when the full use is being made of it at high altitudes.

A supercharger capable of maintaining ground-level power to 12,000 ft. in a 500 h.p. engine would absorb 30–40 h.p. at the ground, and to compensate for this loss of power it is usual to allow the induction-pipe pressure to rise to 1 or 2 lb. per sq. in. above the external pressure for a short time while taking off and climbing from the ground, although the engine cylinders may not be capable of dealing with the extra heat-flow involved for more than a few minutes.

Since an increase of engine speed increases the fan speed and therefore, up to a point, the supercharge effect, the engine can be readily overloaded by allowing its speed to increase. Its well-being and general reliability are therefore at the mercy of the pilot unless some form of automatic control be provided to ensure that a certain maximum pressure for the air delivered to the engine cannot be exceeded under any conditions.

In the matter of control below the full supercharged height the Root's type of compressor has some advantage over the centrifugal type, for it can be by-passed, so that although always driven by the engine its air supply can be diverted from the cylinders when not required, and the power absorbed in driving it can be reduced when the extra air is not wanted. To prevent fuel wastage by such by-passing, however, the carburettor must be placed between the compressor and the engine, and this entails a special pressure-balance system in the carburettor and some increase of fire risk.

ART. 58. *The elementary theory of superchargers.*

The theory of the gear-driven centrifugal supercharger will be fully treated in art. 60, and for the present it will only be touched upon to an extent sufficient to show the characteristics of the type as compared with those of others.

One important characteristic of an aero-engine supercharger is the power required to drive it, per lb. of air delivered per sec. after it has been compressed through a given ratio. The minimum power would be taken by a frictionless compressor of the cylinder-and-piston type, working so slowly that the compression was isothermal. In such conditions the work done per lb. of air would be represented by the area ABC_1D in fig. 93 and would be given by

$$W = P_1 V_1 \log_e \frac{P_2}{P_1} \text{ ft. lb.,} \qquad (44)$$

where P_1 and V_1 represent the intake conditions, and P_2 is the delivery pressure in lb. per sq. ft. P_2 may for the present be assumed to be the pressure of the standard atmosphere at ground-level.

If the compression were adiabatic, no heat being generated by

eddying or friction during the process, then the work done per lb. of air delivered would be represented by the area ABC_2D in fig. 93, and would be given by

$$W = \frac{\gamma}{\gamma-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} - 1 \right] \text{ ft. lb.} \qquad (45)$$

If the compression be other than adiabatic, the compression line BC will be of the form $PV^n = \text{const.}$, in which n may be greater or less than γ, according to whether the balance of heat retained in the gas during compression is greater or less than under adiabatic conditions. In all centrifugal compressors heat is generated by eddying

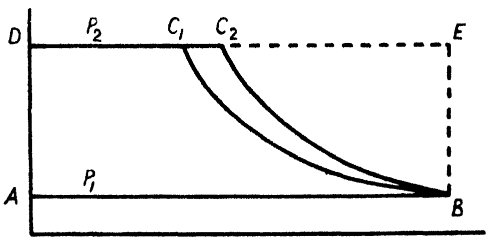

FIG. 93. Graphic representation of the work done by a supercharger in compressing and delivering 1 lb. of air.

and friction, so that the rise of temperature is greater than in the adiabatic case, and the equivalent n is greater than γ.

The action of the impellers in a Root's type of compressor is to transfer a fixed quantity of air, equal in volume to the space B in fig. 91, while it is at the pressure P_1, from the inlet side to the delivery side. The graphic representation of the work per lb. of air necessary for compressing in this way is shown by the area $ABED$ within the dotted lines in fig. 93, for in compressing 1 lb. of air, the rotors have to displace a volume practically equal to the full volume per lb. at the pressure P_1 against the full pressure difference P_2-P_1, and the work done is therefore

$$W = (P_2-P_1)V_1 \text{ ft. lb.} \qquad (46)$$

It is clear from the diagram that for small values of the pressure ratio P_2/P_1 the extra work to be done by the Root's compressor is small, but that it rapidly increases as the ratio of compression becomes greater.

The ratio of the areas $ABC_2D : ABED$ may be called the 'type efficiency' of the Root's compressor relative to the adiabatic process, and its relation to the pressure ratio $R = P_2/P_1$ can easily be shown to be

$$\frac{\gamma}{\gamma-1}\left[\frac{R^{(\gamma-1)/\gamma}-1}{R-1}\right].$$

Taking $\gamma = 1{\cdot}40$, the curve of type efficiency against compression ratio is that shown by the dotted curve in fig. 96.

It is impossible to represent the process of compression in a centrifugal supercharger upon any pressure-volume diagram,* and the efficiency and the work done per lb. of air must be discussed in terms of the changes of temperature which occur.

When considering compression at any height above ground-level, the intake temperature T_1 will be taken as the temperature at that height in the standard atmosphere. The delivery temperature T_2 will, of course, depend upon the efficiency of the compression. We may neglect any difference between the kinetic energy of the air on the entry and the delivery sides of the compressor, which is relatively unimportant, and assume that no heat is lost to the walls while the air is passing through; then in all circumstances the work done per lb. of air delivered is given by

$$W = K_p(T_2 - T_1), \tag{47}$$

in which K_p is the specific heat of the air at constant pressure.

There is a point here over which confusion is apt to arise. Provided there is no interchange of heat between the supercharger casing and the air while it is passing through it, the above expression for the work done per lb. of air delivered *always* holds good. It will be shown in art. 60 to follow simply from equating the energies before and after compression. Under all circumstances, also, the initial and final temperatures and pressures are related by the equation

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n} \tag{48}$$

The less efficient the compression and the greater the temperature rise and the work done per lb. of air, the greater will be the values of n and of T_2. The ratio between the temperature rise which would be produced by adiabatic compression and the rise in an actual compressor, when the pressure ratio is the same, may be used as a measure of the compressor efficiency; for it is also the ratio of the amounts of work done in the two cases. This ratio is called the 'adiabatic temperature efficiency'.

It must be emphasized that when heat is generated within the compressor by friction and eddying, as in a real compressor, the work expended per lb. of air passing can *not* be calculated from equation (45) with the correct value of n substituted for γ. That would give the work per lb. if the amount of heat, H, which is really generated

* Although the initial and final states of the air can be represented by two isolated points on such a diagram.

within the casing, had been supplied by conduction through it, from outside. In that case W would no longer equal $K_p(T_2-T_1)$, but $K_p(T_2-T_1)-H$.

In practice the heat interchange through the casing is probably very small, because the casing is warmed by its proximity to the engine, and W, the work per lb. of air, may safely be taken as equal to $K_p(T_2-T_1)$.

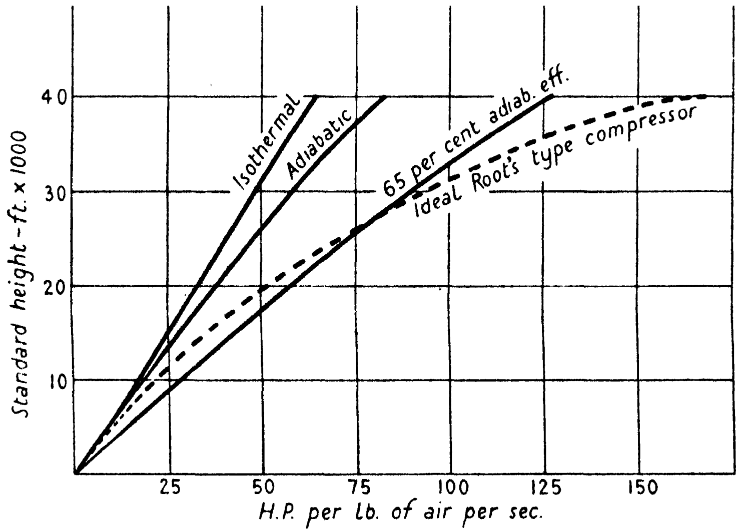

FIG. 94. Horse-power required to compress and deliver 1 lb. of air per sec. at different heights in the standard atmosphere, according to various conditions of compression.

In order to give some concrete idea of the differences in the work per lb. of air when compressing in various ways, curves have been plotted in fig. 94 which show the h.p. necessary to compress 1 lb. of air per sec. and deliver it at normal ground-level pressure at all heights from 0 to 40,000 ft. The curves correspond to compression (*a*) isothermally, (*b*) adiabatically, (*c*) in a compressor of adiabatic efficiency 65 per cent., (*d*) in a perfect compressor of the Root's type.

The figures of h.p. and of the temperature of the air delivered at 10, 20, 30, and 40 thousand ft. are given in table 41. The initial temperatures given in column 2 of the table have been taken as those appropriate to each height in the standard atmosphere.

A rate of supply of air equal to 1 lb. per sec. would suffice for a well-tuned engine of 540 B.H.P. at ground-level. The table shows that to provide this amount of air at 10,000 ft. by means of a geared centrifugal supercharger would absorb 28·5 h.p., apart from the mechanical losses in the gears which would amount to another 5 or

TABLE 41

Delivery temperature and h.p. required to compress 1 *lb. of air per sec. at various heights.*

Standard height ft. × 1000	Initial temp. °C. abs.	Delivery temperature and h.p. required to compress and deliver 1 lb. of air per sec.							
		Isothermally		Adiabatically		In centrifugal supercharger of 65 per cent. efficiency		In a Root's type supercharger	
		Temp.	h.p.	Temp.	h.p.	Temp.	h.p.	Temp.	h.p.
0	288	288	..	..	..	..	..	..	..
10	268	268	17·5	299	18·5	315	28·5	303	21·25
20	248	248	33·7	311	37·8	343	58·2	332	51·0
30	228	228	48·5	325	58·1	375	89·5	383	94·5
40	218	218	64·2	355	82·7	426	127	492	167

6 h.p. Compared with this, a Root's type of compressor in which there were no losses due to eddying, friction, or leakage, could do the same thing for an expenditure of 21·25 h.p.; but at greater heights the essential inefficiency of even the ideal Root's compressor makes the work absorbed by it the greater of the two. In the next article the results of tests will show by how much the performance of the Root's compressor falls short of the curve of type efficiency given in fig. 96.

ART. 59. *The experimental characteristics of superchargers. Root's and vane types.*

An account of the actual performance of centrifugal superchargers will be deferred until after the theory of them has been treated in the next article, and for the present only the experimental results upon the Root's and vane types will be given. Taking the plain displacement, or Root's, type first, the power to drive this with a given pressure difference between intake and delivery depends only upon the displacement and the speed of revolution, and is directly proportional to each. The h.p. per lb. of air per sec. will therefore decrease in proportion to the density of the intake air.

In fig. 94 the h.p. per lb. per sec. was shown for compression at a number of altitudes, and in deriving that curve the temperature and density of the indrawn air was taken to be that in the standard atmosphere at each altitude. In experimental tests of superchargers the pressure conditions of actual service can be reproduced, but not as a rule the temperature conditions. In the experiments of which the results are given below the supercharger was made to compress air in ratios up to 2 : 1, the delivery pressure being always atmo-

spheric. The intake temperature, instead of falling with the intake pressure, remained approximately constant at 15° C.

In fig. 95 the actually measured h.p. per lb. per sec. of air delivered at 3,200 r.p.m. and at different pressure ratios are shown by the points on the curve *AA*.[50] The curve *BB* is the same as a part of the Root's compressor curve in fig. 94, but drawn to a more open scale. The observations of the curve *AA* were made with a constant

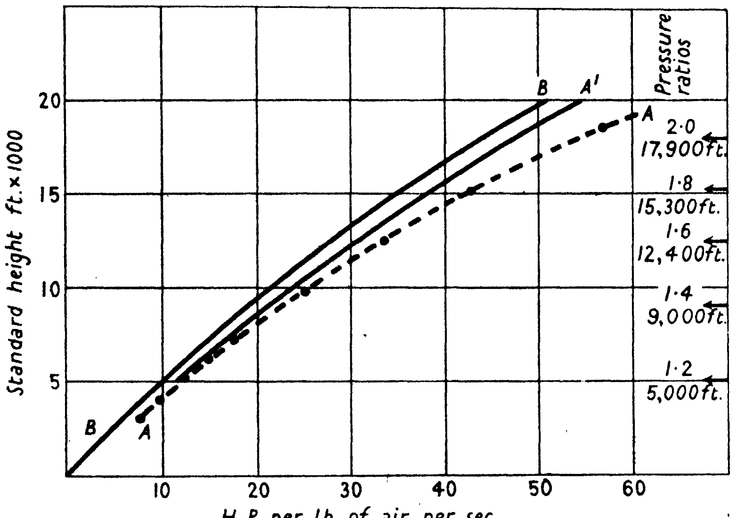

FIG. 95. H.P. required to compress and deliver 1 lb. of air per sec. at various heights in a Root's supercharger.

inlet air temperature of 15° C. When allowance has been made for the lower temperatures and higher densities at heights in the standard atmosphere, to correspond with the curve *BB*, we arrive at the curve *AA'* as representing what the experimental performance would have been at the heights shown in the figure. The horizontal distances between the curves *BB* and *AA'* represent a roughly constant loss due to mechanical and air friction, and leakage.

The speed of 3,200 r.p.m. has been chosen from a series of speeds as typical of a compressor of which the normal speed range would be 3,000–4,000 r.p.m. The h.p. per lb. of air delivered will vary but little with the speed, for both the net power and the rate of delivery are closely proportional to the speed, and the mechanical efficiency is hardly affected. There will be some increase in the heat generated by friction at higher speeds, and some decrease in the leakage back to the low-pressure side per lb. of air delivered; but both effects are

small and the power absorbed per lb. of air delivered may be taken as substantially independent of the speed.

Leakage takes place through the clearances between the two impellers and the casing, and an increase of leakage shows itself as a decrease of the volumetric efficiency—defined as the ratio between the volume of air actually displaced per revolution (measured under the conditions at the intake) and the geometrical displacement volume of the compressor. To a first approximation the rate of leak may be taken as proportional to the clearances, and to work with the minimum possible clearance is of vital importance for a high volumetric efficiency and a minimum power absorption per lb. of air delivered.

It is not possible to maintain the same clearances under different working conditions owing to differences of temperature, and the necessary provision for maximum load may leave the clearances under other conditions sufficient to introduce an appreciable loss. The clearances are at the tips of the rotors, and at each end, and table 42 shows the effect of increasing each of these by about

TABLE 42

Volumetric efficiency of Root's compressor, as affected by impeller clearance. Impeller length 8¼ in. and maximum dimension of cross-section 4 in. Speed 3,200 r.p.m.

	Volumetric efficiency	
Pressure ratio	0·007 *in. tip clearance* 0·020 *in. total end clearance* (*considered impracticable*)	0·010 *in. tip clearance* 0·025 *in. total end clearance* (*practicable*)
1·15	94·4	92·6
1·36	93·0	90·7
1·67	90·9	88·1

0·003 in. in a Root's compressor of which the rotors were 8¼ in. long and 4 in. from end to end of the cross-section, giving a geometrical displacement of 0·382 cu. ft. per impeller revolution. Broadly, the effect of this increase of clearance may be stated as a reduction of the volumetric efficiency by 2 per cent. at a pressure ratio of 1·15, rising to 3 per cent. at 1·67, and the effect was found to be the same over a range of speeds between 2,800 and 4,000 r.p.m. If the compressor capacity be increased by lengthening the rotors, the effect of end clearance can be diminished to some extent, but not that of tip clearance. This type of compressor has been built for aero-engines in three sizes, of which table 42 refers to the intermediate one. In the other two sizes the rotor lengths are 5 in. and 11 in. Tests of the

smaller size under conditions comparable with those of column 3 of table 42 showed a volumetric efficiency varying from 87·5 to 82 per cent., and down to 77 per cent. at a pressure ratio of 2. At the lower speed of 2,800 r.p.m. the fall was greater, from 86·5 to 74·5 for the same range of pressure ratios. In all tests there was a small increase in the volumetric efficiency between 2,800 and 4,000 r.p.m. amounting to between 2 and 3 per cent. caused by the reduction of the time available for leakage back to the low-pressure side.

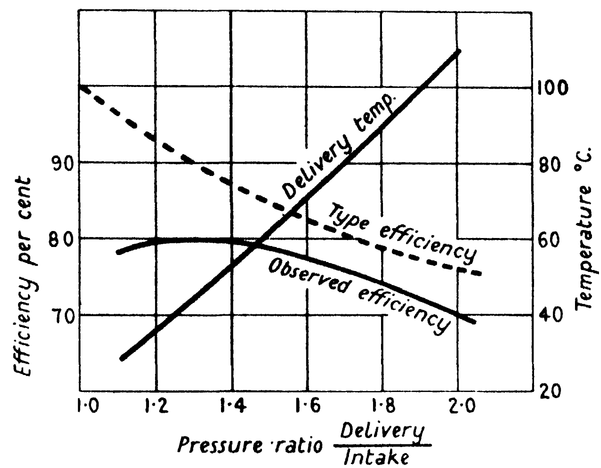

FIG. 96. Adiabatic temperature efficiency of a Root's compressor for different pressure ratios. Delivery pressure atmospheric. Intake temp. 15° C. Speed 3,200 r.p.m.

As regards the mechanical efficiency, this was found to vary between 82 and 92 per cent. according to the speed and the load. It showed no consistent variation with speed, although there were differences probably caused by increased vibration under certain conditions. The chief variation of mechanical efficiency was due, as would be expected, to the different loads on the gears and bearings when delivering against different pressure ratios.

It is possible to calculate an 'adiabatic temperature efficiency' for the Root's supercharger if the temperatures at entry and exit are measured, as well as the pressure ratio. For the tests referred to in fig. 95 this efficiency has been plotted in fig. 96, as well as the delivery temperatures, for the same series of pressure ratios. The efficiency is the ratio between the temperature rise for an adiabatic compression in a certain ratio and the actually measured rise under those conditions. It will be seen that it varied between 80 per cent. and 70 per cent., with a maximum value at the ratio 1·3. This is higher than can be shown by any centrifugal supercharger, as will

appear later, the lower efficiency of the latter type being due to the large eddy and friction losses which accompany the very high speed of operation.

A comparison has been given in a recent American report[51] between a Root's and a Powerplus supercharger, each designed for an aero-engine. The Root's design was the largest of the three sizes already mentioned, with rotors of length 11 in. and a displacement 0·509 cu. ft. per impeller revolution. The test results on this supercharger were taken from an earlier report[52] and corrected to the

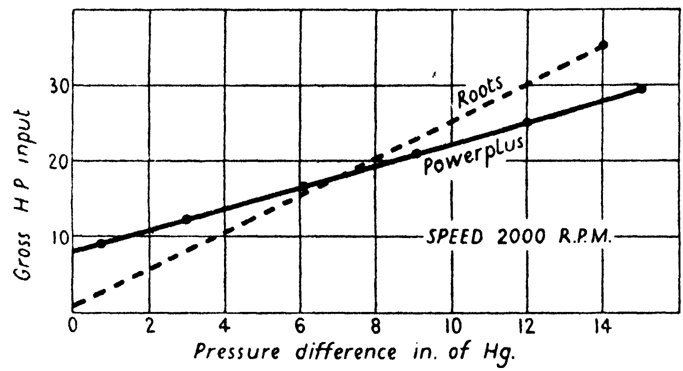

FIG. 97. H.P. required to drive Root's and Powerplus superchargers at various pressure ratios.

same displacement per rev. as the Powerplus supercharger, namely 0·544 cu. ft. per rev. Other leading particulars of this compressor are,

Fixed compression ratio (by vol.)	1·53
Max. speed of drum	3,000 r.p.m.
Weight	119 lb.

In fig. 97 is shown the gross h.p. input for each supercharger at 2,000 r.p.m. and at pressure differences from 0 to 15 in. of mercury, the conditions of test being, as before, that the delivery was always at atmospheric pressure. The range of pressure ratios covered is the same as in fig. 96; but in this case the power is plotted against pressure differences to bring out the linear relationship between them. The speed of 2,000 r.p.m. is chosen as giving results typical of the whole series of tests, which ranged from 500 to 2,500 r.p.m. It may be well to mention that it was found impossible to run the Powerplus machine at more than 2,500 r.p.m. without risk of seizure, whereas the Root's could be worked safely up to an impeller speed of 7,000 r.p.m. if the clearances were suitably adjusted.

Two points are at once evident from fig. 97. In the first place the power to drive the Powerplus at zero pressure difference was nearly eight times that for the Root's; but on the other hand the rise of the power input with the pressure ratio was much less rapid. The high power to drive the Powerplus at zero pressure difference was due in part to a greater mechanical loss, but still more to the fact of its having a fixed compression ratio of 1·53. The effect of this was to

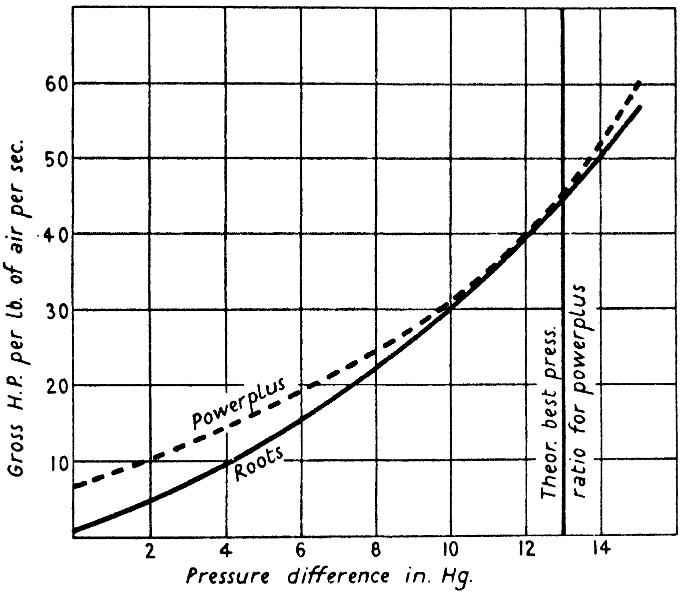

FIG. 98. H.P. per lb. air per sec. required for Root's and Powerplus superchargers.

make it thermodynamically an efficient compressor for a pressure ratio in the neighbourhood of

$$(1{\cdot}53)^{1{\cdot}4} = 1{\cdot}81$$

but extremely inefficient for low pressure ratios; for then work has to be done in raising the air to nearly twice the intake pressure, most of which is pure waste. By shortening or extending the tip D in fig. 90 the ratio of compression within the casing can be lowered or raised, and with it the efficient pressure ratio between inlet and delivery.

The straight lines in fig. 97 do not tell very much about the characteristics of the respective compressors, because the volumetric efficiency and the number of pounds of air pumped per sec. will vary with the pressure ratio differently in the two compressors. Any change of volumetric efficiency is taken account of in fig. 98 which shows the h.p. per lb. of air per sec. at various pressure ratios. The

curve for the Root's compressor will be found to be very closely the same as the curve in fig. 95, in spite of the fact that the tests were upon machines of different size, the one in England and the other in the U.S.A.

The result of the low mechanical efficiency and of the wasted work of compression in the Powerplus compressor is seen on the left of the diagram, at low pressure ratios. The increase of its compressing efficiency up to a pressure ratio of about 1·8 brings down the gross power required until it is just equal to that of the Root's at a pressure difference of 11·8 in. of mercury, corresponding to a ratio of 1·65. Above that point the compressing efficiency soon begins to fall off, and moreover the mechanical friction loss must increase seriously in the Powerplus at large pressure differences. The result of these two factors is that the Root's compressor again shows to advantage in terms of h.p. per lb. of air per sec.

ART. 60. *The theory of the centrifugal supercharger.*

In following the theory of the centrifugal compressor it is well to keep in mind certain important quantities which occur, and to have some general notions about them. In the first place, then, an impeller of a certain size, rotating at a definite speed, is capable of doing a certain definite amount of work in compressing, and giving velocity to, each unit mass of air as it flows through it. This 'work capacity' of the impeller is independent of the throughput or mass-flow of air per sec.* If one imagines the compressor to be delivering through a throttle valve, then an increased mass-flow, following an opening of the throttle, will mean that if the impeller speed is kept constant the driving torque necessary to keep it so will go up in proportion to the mass-flow, and so will the total work done; but the work per unit mass of air remains the same, being dependent only on the impeller speed. In an ideal compressor, moreover, the ratio of compression $R = P_2/P_1$ would not be affected by a change of the mass-flow. It is dependent purely on the work capacity of the impeller and the initial temperature of the air.

In the ideal compressor, again, the constant compression ratio R (for a constant speed) would be associated with the normal rise of temperature for an adiabatic compression, equal to T_2-T_1, where $T_2/T_1 = R^{(\gamma-1)/\gamma}$. This rise of temperature would represent the work done in compressing and delivering each lb. of air, and, if there were no mechanical friction losses, this would be equal to the work necessary for driving the supercharger per lb. of air delivered.

* The impeller is assumed to be of the simple form with radial vanes shown in fig. 92. If it were not, the above statement would need qualification.

In actual compressors:

(1) The 'work-capacity' for a given impeller speed is less than in the ideal case.

(2) The compression ratio falls off as the mass-flow increases.

(3) The temperature rise for a given compression ratio will be greater than $R^{(\gamma-1)/\gamma}$ on account of skin friction between the air and the metal surfaces, and eddy friction within the air itself derived from the breakdown of the ideal streamline flow.

Considering, first, the ideal compressor, let

P_1 = the pressure at entry to the impeller, in lb. per sq. ft.

P_2 and T_2 = the pressure and temperature of the air as finally delivered.

P_0 and T_0 = the atmospheric pressure and temperature. (In the ideal, but not in the real, compressor $P_0 = P_1$.)

$R = P_2/P_0$.

W = work done on each unit mass of air passing, in ft. lb. per sec. per slug.

v_1 and v_2 = the velocities of the air entering and leaving the compressor.

r = the maximum radius of an impeller blade.

U = the impeller blade-tip velocity in ft. per sec.

u = the mean tangential velocity of the air leaving the impeller.

K_p = the specific heat of air at constant pressure, in ft. lb. per slug per deg. C.

From equating the energy of the air at entry and exit from the compressor, we have

$$\tfrac{1}{2}v_2^2+K_pT_2 = \tfrac{1}{2}v_1^2+K_pT_1+W,$$

whence
$$T_2-T_1 = \frac{W}{K_p}+\frac{1}{2K_p}(v_1^2-v_2^2). \tag{49}$$

When evaluated numerically for a practical case, in which T_2-T_1 would be of the order of 50° C., it will be found that the kinetic energy term in this equation amounts to less than 0·5 per cent. of W/K_p, for v_1 and v_2 are of the order of a few hundred ft. per sec. We therefore have the simple relationship already given in equation (47), namely,

$$T_2-T_1 = \frac{W}{K_p},$$

for the inevitable rise of temperature due to the work done upon the air while it is passing through the compressor, provided only that

heat is not lost to the impeller casing during the process. It may be emphasized again that, T_1 and T_2 being the initial and final temperatures, this equation *always* holds, however much heat may have been generated in the air by friction in the compressor. In the ideal compressor, as already stated,

$$\frac{T_2}{T_1} = R^{(\gamma-1)/\gamma}.$$

The value of W may be related to the velocities of the air and impeller blades by considering the momentum of the air round the impeller axis, for the driving torque Q equals the change of momentum per sec. and therefore, per slug of air flowing,

$$W = Q\omega = ru\omega = uU. \tag{50}$$

or, if there are fixed guide vanes at the air intake which give to it a tangential velocity u' at a radius where the impeller blade velocity is U', then

$$W = uU - u'U'. \tag{51}$$

In aero-engine superchargers the term $u'U'$ is never likely to be important in comparison with uU, and for the sake of simplicity it will be neglected in what follows. It will also be assumed that the impeller has straight radial blades. Experiments have been made from time to time with impellers of curved blade form, but at the extreme speeds of rotation required in aero-engine superchargers it has not been found that the additional difficulty and cost of manufacture was justified by the results obtained, and blades of the simpler form with radial tips are almost universally used.

So long as the blades are radial at the tips, the maximum possible work capacity of the impeller is given by

$$W = U^2, \tag{52}$$

and this value would be obtained if the air could be made to leave the blades with a *mean* tangential velocity, u, equal to that of the blade tip. In practice the value of u is always less than U, and varies between about 98 and 85 per cent. of it, according to the mass-flow of air through the supercharger. With a frictionless fluid, u would be independent of the mass-flow; but even so it would always be less than U so long as the blades were radial, for reasons explained below. With curved blades, of course, u could be made equal to, or greater than, U by giving the fluid a component of tangential velocity relative to the blade, before it left the tip. In these circumstances the work capacity in a frictionless compressor would clearly be no longer independent of the mass-flow in general, although it may be noted in passing that when intake guide-vanes

are fitted there is a particular impeller form (not radial) which would make it so. The analysis of these special cases, however, is beyond the scope of this book.

Before passing on to consider the real compressor with fluid friction present, and its relation to the ideal one, an attempt must be made to describe the flow conditions within it, and to relate these, and the losses involved, to the known facts about fluid

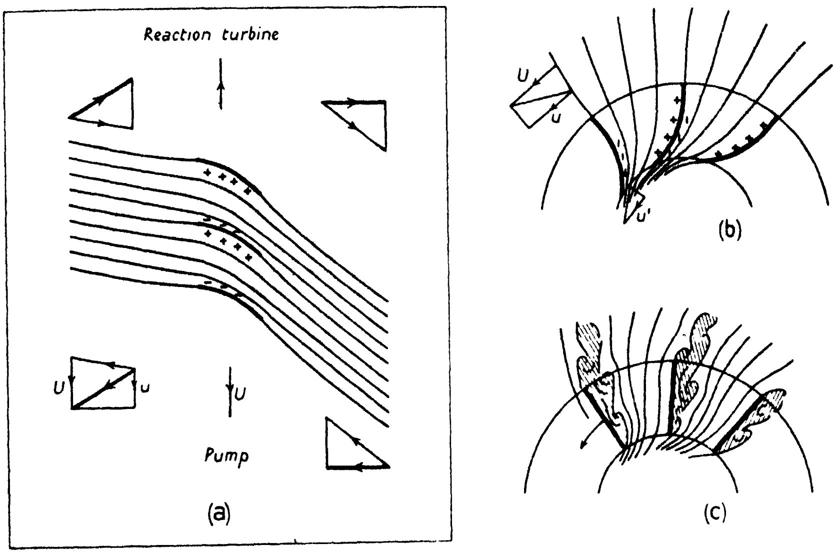

FIG. 99. Fluid flow between guide-vanes: (*a*) axial flow, (*b*) and (*c*) radial flow.

flow in general as they have been observed, for example, in wind tunnels.

The essential features of the process of changing the direction, speed, and pressure of a uniform stream of frictionless fluid are shown in fig. 99 (*a*). Across the stream is placed a row of curved blades or aerofoils shaped so as to produce a reaction on the fluid in the direction of the desired change in momentum. The blades themselves experience an equal and opposite reaction, analagous to the lift of an aeroplane wing. Associated with this reaction is a difference of pressure between the concave (high pressure) and convex (low pressure) sides of each blade; and in accordance with Bernoulli's equation, whereby high pressure is associated with low speed, and vice versa, it follows that between the blades the stream-lines are not parallel, but are as shown in the figure (high pressure regions denoted by +, low pressure by −). The mean angle, δ, between the initial and final direction of the stream must therefore

be less than the maximum change of direction, δ', of the stream-lines which pass close to the blades.

When the row of blades is so arranged (with its general direction bisecting the angle between the incoming and outgoing streams) that the effective area available for flow is unchanged, both the speed and the pressure of the fluid will be unaffected by the change in direction. When, however, the arrangement is unsymmetrical, as in fig. 99 (*a*), there will be a change of speed, and a consequent change of pressure, in accordance with Bernoulli's equation. If, in fig. 99 (*a*), the flow is from right to left, there will be a fall in speed and a rise in pressure; if from left to right, a rise in speed and a fall in pressure.

By superposing vectorially on the whole picture a velocity U parallel to the row of blades we arrive at the impulse turbine (symmetrical arrangement), the reaction turbine, and the pump, with *axial* flow of the fluid in each case. In fig. 99 (*a*) the arrows show, for the last two, the appropriate directions of the blade and fluid velocities, absolute fluid velocities being indicated by thick lines. It will be seen that conditions have been so arranged that in the pump the absolute velocity of the fluid entering, and the tangent to the edges of the blades where the fluid leaves, are at right angles to U. Since, however, the *mean* velocity of the fluid leaving, relative to the blades, is inclined to U at an angle greater than 90°, the tangential component, u, of the absolute velocity of the fluid leaving is less than U. This must necessarily be true for any finite number of blades.

The blades of the axial flow pump of fig. 99 (*a*) are suitable for one ratio only of the axial velocity of the fluid to the blade velocity, for in a real fluid the flow would leave the blade on one side or the other of the sharp entering edge at any but the correct ratio, and turbulence would at once arise. A rounded entering edge, as used on aeroplane wings, would permit some latitude in the ratio, for then the blades would become thin streamline forms round a bent centre line, and there would be a range of angles of incidence for which the fluid flow-lines would follow the surface. In a real fluid there will, however, be further risk of separation of the flow from the blade owing to slowing up of the frictional boundary layer due to the rising pressure along the convex side of the blade, towards the point where the fluid leaves it. The larger the pressure difference between the two sides of the blade, the more likely is this to occur. In other words, the blades will 'stall' if they are overloaded, through being too few in number or too small in surface for the work they have to do.

A similar synthesis of the radial flow pump leads to generally

similar conclusions, with some important modifications. In fig. 99 (*b*) an impeller with radial blade tips is shown, together with the corresponding (relative) stream-lines. Superposition of a uniform angular velocity leads to the vector diagrams shown for the *mean* flow, and once again the tangential component, u, of the absolute velocity of the fluid is less than the peripheral speed of the blade tips, U. The absolute tangential component at entry, u', is essentially zero, in the absence of fixed guide blades. Moving blades curved forwards at entry would be necessary to avoid turbulence there, if the flow were radial, but in practice it cannot be purely radial and is often more nearly axial, suggesting a blade whose form at entry combines some features of both fig. 99 (*a*) and 99 (*b*).

The general divergence of the passages between the blades owing to the radial arrangement, suggests a more rapid rise of pressure than in the axial flow arrangement of fig. 99 (*a*), and hence an increased tendency towards turbulent flow in the critical region. It must be remembered, however, that the actual impeller is not of uniform axial width, and there is evidence that, in a passage in which expansion in one direction is accompanied by contraction in the perpendicular direction, the flow may be practically free from turbulence. Thus the probability of turbulence arising cannot be judged solely from the appearance of the impeller in two dimensions only, as in fig. 99 (*b*). Nevertheless there is no doubt that the main causes of turbulence in actual impellers are those described above, and that with radial vanes at exit the tangential velocity, u, of the fluid as it leaves them is less than the peripheral speed U of the blade tips, to an extent which depends on the flow through the impeller and is due partly to the fundamental characteristics of the nominal streamline flow and partly to such turbulence as does in fact arise. In fig. 99 (*c*) the general nature of this turbulence for straight radial blades is shown diagrammatically.

The flow between the fixed vanes of the diffuser is subject to the same conditions as those outlined above, and the non-uniformity of the flow from the impeller makes some breakaway at entry to the diffuser vanes inevitable unless they are made of streamline form, even apart from the variation of the mean resultant velocity of flow with changes of mass-flow and impeller speed. As with the impeller, conditions of manufacture preclude any careful shaping of the diffuser vanes, and there is the further consideration that if the diffuser chamber is to be kept reasonably small, no thickening of the vanes into a streamline form is possible without cutting down too much the cross-sectional area of the passages available for the fluid to flow in its spiral path.

Some experimental results will be given in the next article which go to show that the losses in the diffuser, for a constant impeller speed, are very much affected by the mass-flow; as would, indeed, be expected, for this directly affects the radial velocity and the average angle of entry to the diffuser vanes. Anything more than a small divergence from the designed angle must cause breakaway of the boundary layer at the leading edges of the vanes and turbulence through the whole length of the diffuser passages.

Turning, now, to an expression for the efficiency of a real, as compared with the ideal, compressor, let the total ratio of compression be R, and let

$$\frac{T'}{T_1} = R^{(\gamma-1)/\gamma}, \tag{53}$$

then, as before,
$$W = K_p(T_2-T_1), \tag{54}$$

and if W' is the amount of work which would have to be done per unit mass of air in a frictionless compressor to give the same compression ratio,

$$W' = K_p(T'-T_1). \tag{55}$$

The efficiency of the compressor may be defined as

$$\eta = \frac{W'}{W} = \frac{T'-T_1}{T_2-T_1}. \tag{56}$$

This is what is known as the 'adiabatic temperature efficiency'. It expresses the efficiency of the compressor as compared with a similar, but frictionless, one in which no heat is lost or gained by the air during its passage through the casing.

From equations (53) and (55) it follows that

$$R^{(\gamma-1)/\gamma} = 1+\frac{\eta W}{K_p T_1}. \tag{57}$$

This equation shows that in a compressor of a definite efficiency the compression ratio will depend only on the work per unit mass of air and its initial temperature. Furthermore the equation

$$W = K_p(T_2-T_1) = u\dot{U}.$$

still holds good when heat is produced by friction. It expresses the fact that under all circumstances the 'work-capacity' depends only on the speed of the impeller tips and the mean tangential velocity of the air as it leaves them. An increase of the mass-flow will mean a proportionate increase of the radial, but not of the tangential, velocity, if the blades are radial at the tips. The equation gives a perfectly general relationship between this mean tangential velocity

u, the impeller tip-speed U, and the total rise of temperature in the compressor.

What, it may be asked, would happen if a frictionless compressor, driven at a certain speed, and therefore with certain fixed values of U, u, and (T_2-T_1), were suddenly to become subject to fluid friction without any change of speed? Must there not be an increase of the rise in temperature through the compressor? To this question it is impossible to give a short and simple answer. To a first approximation the answer is that the compression ratio, and what may be called the 'useful temperature rise', would drop, but the total temperature rise and the work done per lb. of air would remain unaltered.

In passing from the ideal to the real condition, however, there are secondary effects introduced which upset any simple comparison with the ideal supercharger. Free vortices would be set up between the blades, as a result of breakaway of the boundary layer, and these would increase the necessary driving torque and the work capacity for a given impeller speed. The direction of motion of the vortices as they pass through the impeller is such as to increase the pressure on the driving faces of the blades and reduce that on the suction faces. The necessary driving torque for a given impeller speed and mass-flow is increased in consequence, and the additional energy put into the air leaves the impeller as the kinetic energy of the free vortices. These results have been partially confirmed by experiments in which the observed ratio u/U was greater than the calculated value for the ideal case, and was sometimes even greater than unity with radial blades.

An approximate expression for the necessary conditions at the impeller periphery of a real supercharger to give a certain compression ratio follows from equations (50) and (57), for

$$uU = \frac{K_p T_1}{\eta}(R^{(\gamma-1)/\gamma}-1), \tag{58}$$

and on the assumption that $u = U$ this gives a figure for the necessary impeller tip-speed. In actual compressors u generally varies between about 98 and 85 per cent. of U, diminishing with an increase in the mass-flow.

Single-stage compressors in service achieve an adiabatic efficiency of about 0·65, so that if one is required to deliver air at ground-level pressure when at 20,000 ft., where the temperature would be 248° C. abs. and the relative pressure 0·459, the necessary tip-speed would have to be

$$\left[\frac{10{,}570\times 248}{0{\cdot}65}(2{\cdot}18^{0{\cdot}286}-1)\right]^{\frac{1}{2}} = 1{,}000 \text{ ft. per sec.}$$

With an impeller of 10 in. diam. this would mean a revolution speed of 23,000 r.p.m. Apart from the mechanical problems of reliable operation at such a speed, the air velocities will be very near to the speed of sound, and the conditions clearly indicate a limit of what could usefully be attempted in a single-stage compressor. Up to the present such compressors have not been employed in general use for maintaining ground-level pressures to heights above about 12,000 ft.

If the air, after leaving the impeller tips, were allowed to form a large free vortex co-axial with the impeller, so that it travelled in a spiral path to exit pipes at the periphery of the casing, it would lose velocity and gain pressure in accordance with the ordinary laws of vortex motion. Diffuser blades, in fact, are unnecessary, except to hasten the transformation of the kinetic energy of the air into pressure energy. The tangential velocity of the air as it leaves the impeller is of the order 800–1,000 ft. per sec., and to reduce this to a few hundred ft. per sec. in a free vortex chamber would mean a casing surrounding the impeller impossibly large for fitting to an aero-engine.

The air after leaving the impeller is usually allowed to form a free vortex out to some radius r', at which point the diffuser vanes begin. Suppose the tangential velocity has fallen to u' at the radius r', then from the laws of vortex motion

$$ur = u'r',$$

and hence

$$u' = \frac{ur}{r'} = \frac{W}{U}\frac{r}{r'} = \frac{W}{\omega r'}. \tag{59}$$

From this equation one may deduce the important fact that if a compressor is to be designed to give a definite compression ratio, that is for a definite value of W, then the larger the angular velocity ω of the impeller, the smaller will be r', the radius at which the tangential velocity can be reduced by a free vortex to some predetermined value u'. In other words, the most compact supercharger will be obtained by using the highest possible angular velocity of impeller, irrespective of its radius.

The same method of treatment may be applied to the whole compressor, including the section which includes the diffuser blades. In that case u' becomes the air velocity at exit; r', however, must be interpreted not as the actual maximum radius, but as the latter increased by the presence of the diffuser blades to what may be called the 'effective radius' for the desired exit velocity.

The complete process of compression from P_1 to P_2 is divided be-

tween the impeller and the diffuser in a proportion which depends upon the relation between U and u.

Considering flow through the impeller only, and neglecting radial as compared with tangential velocity, we have, if T be the temperature of the air as it leaves the blade tips,

$$K_p T+\tfrac{1}{2}u^2 = K_p T_1+W = K_p T_1+Uu. \qquad (60)$$

It should be noted that $\frac{1}{2}u^2$ is not strictly correct in this equation as an expression for the kinetic energy, which depends upon the mean square velocity, because u was defined as the mean velocity, and the square of the mean velocity is less than the mean square in a periodic flow, such as must be occurring at the exit from an impeller with a finite number of blades.

Assuming for present purposes that the error is small, the above equation leads to

$$K_p(T-T_1) = \tfrac{1}{2}U^2-\tfrac{1}{2}(U-u)^2, \qquad (61)$$

which shows that the maximum possible rise of temperature and pressure in the impeller itself would occur when $u = U$, and that the rise of temperature would then be $U^2/2K_p$. In other words, the best that the impeller can do is to communicate its energy W to the air, so that the total amount, U^2/K_p, is equally divided between kinetic and pressure energy. In all compressors with radial blade tips, as we have seen, u will be less than U, and more than half the increase of energy in the air as it leaves the impeller will be in the kinetic form. This is a disadvantage in so far as it will involve higher skin friction and eddy losses in the diffuser.

As already mentioned, correct adjustment of the diffuser vanes is only possible for one impeller speed and rate of flow, and in consequence the most serious dissipation of energy in the form of heat is probably to be associated with entry into the diffuser passages, and equally, perhaps, with entry into the impeller. Other losses occur through leakage (small, however, in this type of compressor) and through the periodic nature of the flow already mentioned. Here the reduction of the loss to a minimum is a question of balancing the eddy loss set up by the periodic flow against the increased skin friction with the impeller blades, if the number of these be increased so as to reduce the periodic nature of the flow. The optimum number of blades is a matter which can only be settled by experiment.

Art. 61. *Experimental characteristics of superchargers.*

The performance of actual superchargers may be conveniently expressed by curves, plotted to a base of mass-flow of air per sec., which show how the compression ratio, temperature rise, and adia-

batic efficiency are affected by the mass-flow at a constant impeller speed. It makes no difference to the form of the curves whether the mass-flow is controlled by operating a throttle valve on the delivery or the inlet side of the compressor; although the range of pressures and mass-flows would normally, of course, be different.

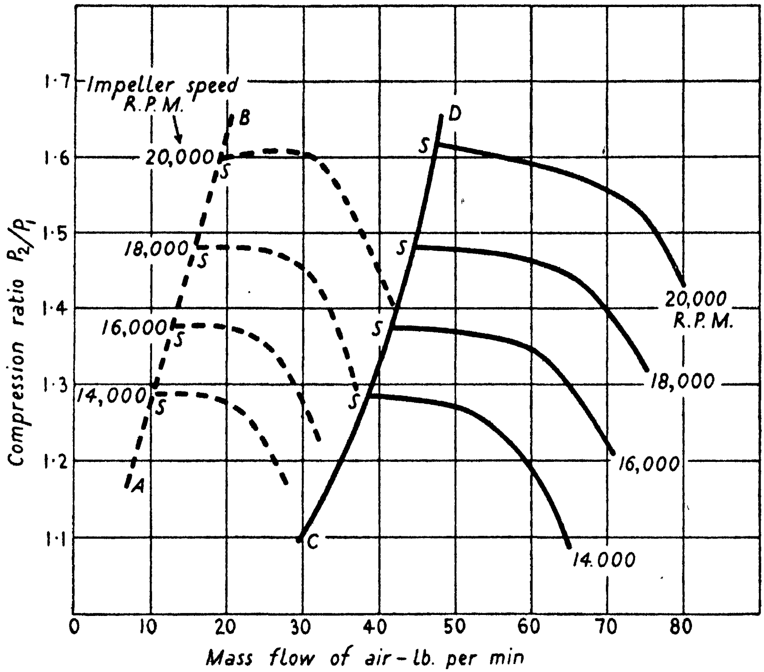

FIG. 100. Characteristic curves for centrifugal supercharger. Impeller of 10·3 in. diam. with unshrouded radial vanes.

Full-line curves = Diffuser with straight vanes at 12°.
Dotted curves = Diffuser of spiral volute form without vanes.

All superchargers show the same general relationship between compression ratio and mass-flow, and the curves are often referred to as the 'characteristic curves' of the centrifugal compressor. Typical characteristic curves for a compressor with plain radial impeller blades are shown in fig. 100.[58] The full-line curves and the dotted curves refer to tests of the same impeller, of diameter 10·3 in., at the same series of rotational speeds, in combination with two different types of diffuser. The full-line curves are for the normal type of diffuser chamber with straight vanes arranged at an angle of 12° with the tangents to the circle on which their inner tips lie; the dotted ones were obtained when a spiral volute receiving cham-

ber, without any vanes, replaced the normal diffuser. It will be noticed that at each impeller speed the maximum compression ratio was substantially the same, but that the dotted curves, besides covering a lower range of mass-flows, all show a certain range of constant compression ratio instead of one which falls at an increasing rate from the lowest mass-flows upwards.

It is a general feature of the characteristic curves of all centrifugal compressors that, if the mass-flow is reduced by throttling while the impeller speed remains constant, there comes a point, known as the 'surge point', beyond which the delivery pressure cannot be made to show any further increase. In superchargers having the usual type of diffuser, with vanes, there is an abrupt *fall* of the delivery pressure when the mass-flow is reduced beyond the surge points S on the full-line curves of fig. 100. Superchargers with the vaneless or spiral volute types of diffuser do not show a sudden drop at the surge point, but no further throttling of the mass-flow can produce any increase of the delivery pressure. It will be seen that at 20,000 r.p.m. the dotted curve in fig. 100 had already begun to show a slow fall of the delivery pressure before the condition of unstable flow was reached, at the point S. This point of break-down in the relationship between mass-flow and delivery pressure (or between mass-flow and inlet pressure when throttling is at that end) is accompanied by a pronounced audible vibration, or surging, in the air-stream. The noise sometimes amounts almost to a scream.

The character and amplitude of the vibration is much influenced by the type of diffuser employed and by the volume of any space between the compressor and its throttle valve, whether this is on the delivery or the inlet side. In general it may be said that with the normal type of diffuser, with guide vanes, there is a sudden drop of compression ratio at the surge point; whereas with the spiral volute or vaneless diffuser the break-down of the compression is less sudden but may be accompanied by violent oscillations of pressure of large amplitude and quite low frequency, 2 or 3 per sec., sometimes sufficient to cause a reversal of the air-flow through the compressor, according to the volume between this and the throttle valve. With the normal diffuser the break in the delivery pressure is accompanied by the characteristic noise, but not by the same oscillations of pressure.

The two limit curves AB and CD in fig. 100, joining up the ends of the characteristic curves, show how the surge points S vary with the speed for each type of diffuser.

Turning now to a more complete analysis of one particular set of conditions, the curves of fig. 101 show the variations of compression ratio, temperature rise, adiabatic efficiency, and total power input to

the impeller, as the mass-flow increased from 24 to 40 lb. per min. The tests were made with the spiral volute diffuser and an impeller with radial vanes.[53]

From equations (47) and (50) one can calculate the horse-power input to the air itself at each mass-flow, and the variation of u, the mean tangential velocity of the air as it left the impeller. The impeller tip-speed was 895 ft. per sec. throughout.

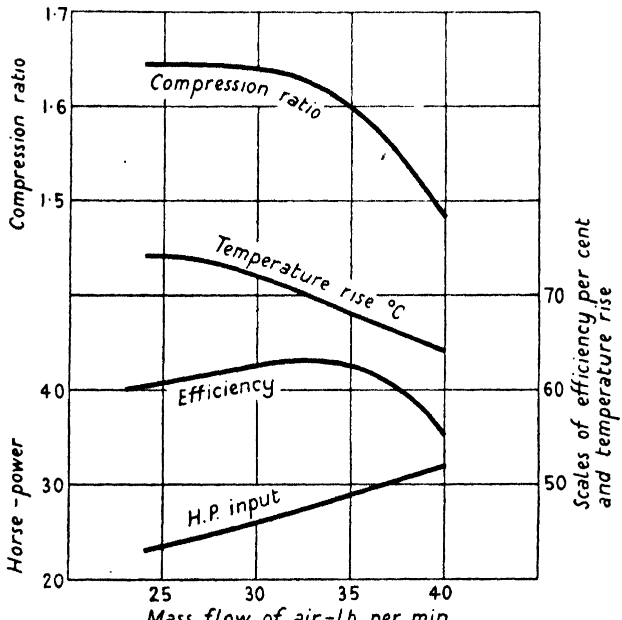

FIG. 101. Experimental data from test of 10·3 in. impeller at 20,000 r.p.m. with spiral volute diffuser.

The difference between the power input to the air and the total power gives the power loss in the gears and bearings. This, together with the variation of u, is shown in fig. 102. There is a very slight increase of mechanical loss with increasing mass-flow, as might be expected from the heavier driving torque. It amounts to an increase of about 0·5 h.p., or 10 per cent., over the range of the experiments.

At the minimum air-flow, 24 lb. per min., the mean tangential velocity is within 20 ft. per sec. of the constant top speed U, but drops steadily to 755 ft. per sec. at the maximum flow of 40 lb. per min.

It is to be noted that the maximum efficiency does not occur at the mass-flow giving maximum compression ratio. What the efficiency is, depends upon how much of the energy put into the air becomes dissipated into heat by eddies and skin friction. This fric-

tion heat is generated partly in the impeller and partly in the diffuser stage.

What the compression ratio is, depends upon the product of the efficiency and the work, W, which the impeller is able to do on each unit mass of air. The compression ratio varies less rapidly than the product, for

$$R^{(\gamma-1)/\gamma} = R^{0\cdot286} = 1+\frac{\eta W}{K_p T_1},$$

and, when R is about 1·5, a 10 per cent. increase in ηW gives a 4 per cent. rise in compression ratio. It should be kept in mind

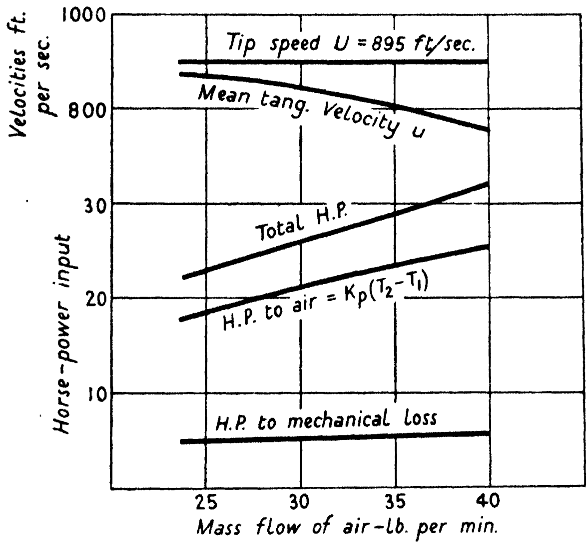

FIG. 102. Experimental data from a test of a centrifugal supercharger with 10·3 in. impeller.

that W (indicated by the total rise of temperature) is not affected by the friction losses. It falls off with increasing mass-flow, as shown by the temperature rise curve in fig. 101, owing to the inability of the impeller to maintain the mean tangential velocity u. This does not represent a loss in the compressor, properly speaking, but a failure to put as much work into each unit mass of air as we should like. The impeller is being maintained at constant r.p.m., and the driving torque necessary to do this for any given mass-flow of air is less than it would be if the value of u could be maintained.

As the mass-flow is increased from the minimum upwards, the value of W steadily falls, while η at first rises and then falls more and more rapidly. As a result of these two variations, R falls, very slowly

at first, but then more and more rapidly as the rapid fall of efficiency becomes the dominant factor.

The efficiency will be a maximum at that particular combination of radial and tangential velocities in the air which happens to produce a minimum total amount of heat from eddy and skin friction. At what mass-flow this occurs will depend upon the angle of the diffuser vanes, if there are any, and upon the design of the impeller, whether shrouded or otherwise. The subdivision of the friction heat between the impeller and diffuser will also vary with the design features of the compressor, but one can say that, in general, the rapid fall of efficiency at high mass-flows is due to the rapidly increasing proportion of friction heat produced during the diffuser stage.

By taking readings of the static pressure, as well as of the maximum value of the pitot pressure $(p+\frac{1}{2}\rho v^2)$ in the air just as it leaves the impeller vanes, it is possible to subdivide the friction losses in impeller and diffuser. For let P and P' be the static and pitot pressures at exit from the impeller, P_1 and T_1, and P_2 and T_2 representing conditions at entry and exit of the compressor, as before.

Then T, the temperature of the air leaving the impeller, can be calculated, for

$$\frac{T_2}{T} = \left(\frac{P'}{P}\right)^{(\gamma-1)/\gamma}. \tag{62}$$

This follows from the fact that $P' = P+\frac{1}{2}\rho v^2$, where v is the resultant velocity of the air as it leaves the impeller, and therefore P' is what P_2, the final pressure, *would* have been, *if* the energy represented by $\frac{1}{2}\rho v^2$ had been converted adiabatically into pressure energy.

Now the work done in compressing the air in the impeller is

$$W_i = K_p(T-T_1). \tag{63}$$

W_i is not of course the whole work done on the air. There is the kinetic energy, also given to the air by the impeller, which suffices for the further work of compression in the diffuser W_D, and

$$W_D = K_p(T_2-T). \tag{64}$$

The total work done by the impeller is, as before,

$$W = W_i+W_D = K_p(T_2-T_1). \tag{65}$$

The work which would have been done in the impeller and diffuser respectively, if the process had been frictionless, would have been

$$W_i' = K_p T_1\left[\left(\frac{P}{P_1}\right)^{(\gamma-1)/\gamma}-1\right] \tag{66}$$

and

$$W_D' = K_p T\left[\left(\frac{P_2}{P}\right)^{(\gamma-1)/\gamma}-1\right]. \tag{67}$$

The losses in the impeller and diffuser, expressed as a fraction of the total work done on the air, are therefore

$$\frac{W_i - W_i'}{W} \quad \text{and} \quad \frac{W_D - W_D'}{W}.$$

In fig. 103 are given the curves of P, P', and P_2 for the same compressor with the spiral volute diffuser, and at the same impeller

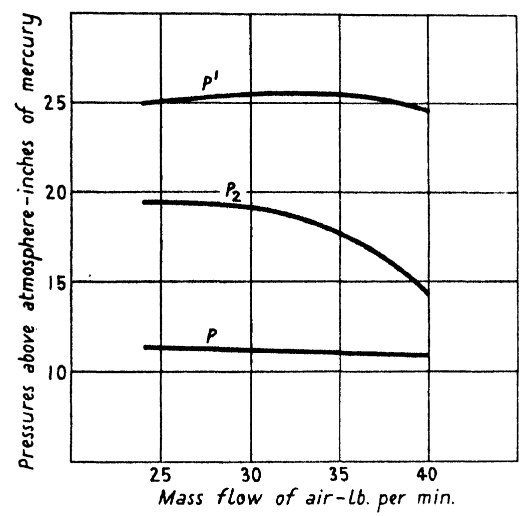

FIG. 103. Experimental data from a test of a centrifugal supercharger with 10·3 in. impeller.

speed, as for the results in figs. 101 and 102. P_1 was normal atmospheric and T_1 15° C. throughout. The temperature rise curve of fig. 101 therefore gives T_2.

All the data are therefore available for calculating the losses according to the method given above. The results are shown by the curves in fig. 104 where the losses are expressed as percentages of the total work done per unit mass of air.

The impeller loss diminished from 22 per cent. to 12 per cent. of the work per unit mass over the range of mass-flows; and the work, as indicated by the temperature rise in fig. 101, itself diminished by some 13 per cent. It follows that according to these results the *total* amount of heat developed by eddies and friction throughout the impeller stage was very substantially less at a flow of 40 than at 25 lb. per min.

This gain in impeller efficiency is balanced by an increasingly rapid rise in the heat produced during the diffuser stage, so that

after a slow increase of overall efficiency up to a flow of about 33 lb. per min., there is a rapid falling away owing to the inefficiency of the action during progress through the diffuser.

A similar subdivision of the losses was not made for the more normal supercharger with vanes in the diffuser; but one may conclude that a comparison between the two types would be somewhat as follows. The rapid increase of the diffuser loss in the spiral volute type, above the comparatively small mass-flow of 35 lb. per min., was

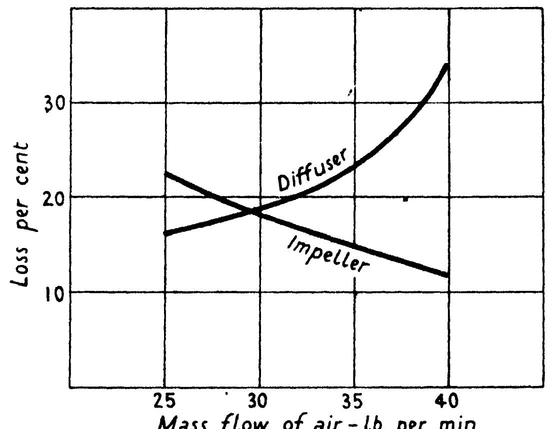

FIG. 104. Eddy and skin friction loss in impeller and spiral volute diffuser, expressed as a percentage of the total work done per unit mass of air. Impeller speed 20,000 r.p.m. Impeller tip speed 895 ft. per sec.

due to the fact that the radius of the volute was not large enough for the efficient conversion of kinetic into pressure energy as a free vortex; and at a flow of 35 lb. per min. or thereabouts violent eddying was set up.

By control of the spiral flow of the air between diffuser vanes after it has left the impeller, as in the other type, the formation of eddies can be much reduced, so that the same compression ratio, and nearly the same overall efficiency, can be produced at nearly three times the mass-flow and therefore at three times the radial velocity of flow through the impeller. At the same time this increased velocity through the impeller probably meant somewhat higher impeller losses.

The fall of the efficiency and compression ratio with an increase of the mass-flow can be seen from fig. 100 to have been less sudden with the vane type than with the spiral volute diffuser. This was probably due in part to the greater importance of the impeller loss,

and in part to the fact that the angle of the diffuser vanes at the inner radius can, at any given speed, be correct at only one value of the mass-flow; and this flow would be arranged to be in the neighbourhood of that giving the maximum compression ratio. It follows that the eddy losses will begin to increase steadily as soon as the optimum flow has been passed, and from this increase of the eddy loss will follow the steady fall of the compression ratio exhibited by the full-line curves of fig. 100.

ART. 62. *The effect of petrol evaporation on the performance of a centrifugal supercharger.*

In discussing the centrifugal supercharger it has been assumed hitherto that the fluid undergoing compression was air only. Although not universal, a common arrangement with aero-engines is to place the carburettor on the suction side of the supercharger, so that in fact what is dealt with by the impeller and diffuser is a mixture of air and partially evaporated petrol. If the petrol, in a 1 : 13 mixture with air, were entirely evaporated before entry to the impeller it would lower the temperature of the mixture by about 25° C., and equation (57) shows that this should increase the compression ratio at a given speed.

The increase would be calculable for the condition of complete evaporation, but in practice this does not usually occur, and it becomes a matter for experiment to determine how far the presence of petrol, introduced as a liquid immediately before entry to the supercharger, affects the pressure ratio at a given speed and mass-flow. The result will clearly depend on the initial temperature of the air and also on the volatility of the petrol, for it is only what is evaporated early that will have any considerable influence on the supercharger performance. If the volatility and air temperature were so low that almost no evaporation occurred within the supercharger casing, then the passage of the liquid droplets swept along with the air, mostly on the metal surfaces, would have little effect.

Until recently experimental data was lacking upon the amount of evaporation to be expected, and upon its effect, but *R. and M.* 1574 has now provided information of much interest. In the experiments described, a centrifugal supercharger of the normal type, having an impeller $10\frac{1}{4}$ in. diam. with ten radial vanes, was supplied with air at three different temperatures, approximately 20°, 40°, and 60° C., and for each inlet air temperature two different rotational speeds were employed. In each of these six sets of conditions experiments were made compressing (*a*) air only, (*b*) air and petrol in the ratio

13 : 1, and (*c*) air and paraffin* in the same ratio. The use of the paraffin was to simulate the conditions when petrol is associated with very cold air. This condition was impossible to reproduce in the laboratory on account of the difficulty from snow formation in the carburettor when air is cooled from ground-level conditions. To have dried the quantity of air taken by the compressor would have required a large special installation. The paraffin distillation curve was very similar to that of the petrol used, but displaced on an average about 85° C. above it, so that to a first approximation the characteristics of the supercharger with the air-paraffin mixture should be similar to those with petrol at an average air temperature 85° C. lower.

A typical set of curves for the high-speed series of tests, showing the effect of evaporation when the fuel is present, are given in fig. 105. The speeds throughout the series are not held quite constant, but are varied in relation to the inlet temperature T_0 so as to maintain constant the factor $N/\sqrt{T_0}$. The abscissae, likewise, are not simply the mass-flow of air W lb. per min. but the factor $W/N\rho_0$, where ρ_0 is the initial density of the air at entry to the carburettor. Any one of the curves in fig. 105 shows the relation between the pressure ratio P_2/P_1 and the mass-flow W for constant values of N, T_0, and ρ_0 and, furthermore, the fact that for different values of T_0 the factor $N/\sqrt{T_0}$ was unaltered, makes the pressure ratios shown on the different curves strictly comparable.

For a discussion of the dimensional treatment of the many variables which enter into the performance of a centrifugal supercharger *R. and M.* 1336 should be consulted, and all that need be said here is to point out that any one variable, in this case the pressure ratio, can be shown to be a function of two non-dimensional factors only, if these are suitably chosen from among the seven independent variables that enter into the performance. In fig. 105 the non-dimensional factors chosen are $N/\sqrt{T_0}$ and $W/N\rho_0$.† Since the pressure ratio is a unique function of these two, the same curve of pressure-ratio plotted against one of them ($W/N\rho_0$) is obtained, so long as the other ($N/\sqrt{T_0}$) is maintained constant, no matter how much N and T_0 may vary individually. In the experiments illustrated by the three coincident curves at the top of fig. 105, T_0 varied from 293 to 333 and N from 18,150 to 19,350 r.p.m., but $N/\sqrt{T_0}$ was very nearly 1,060 throughout.

With the air-petrol mixture the raising of the pressure ratio was about 0·05, from 1·45 to 1·50 at $W/N\rho_0 = 0{\cdot}045$ even when $t_0 = 20$,

* Of the type known as 'white spirit' with a boiling range between 155° and 240° C.

† Since N is to be proportional to $\sqrt{T_0}$, $W/N\rho_0$ is equivalent to $W/\sqrt{(p_0\rho_0)}$.

and was just twice that when $t_0 = 60°$ C. With paraffin, on the other hand, the amount evaporated at $t_0 = 20°$ C. was so slight as to produce on an average no change of the pressure ratio. An interesting point is that the fall of the pressure ratio with an increase in the mass-flow of the paraffin mixture was considerably more rapid than with air only, so that at values of $W/N\rho_0$ above 0·05 the pressure ratio was lower; a fact perhaps to be explained by the increased skin

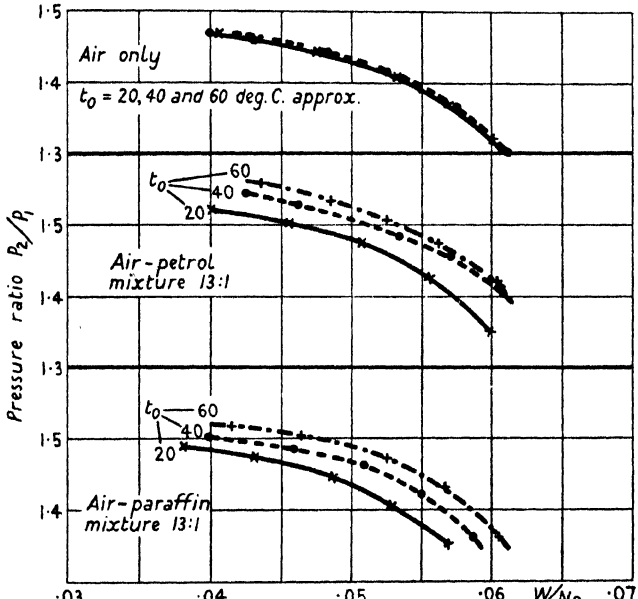

FIG. 105. Effect of fuel evaporation on the pressure ratio of a centrifugal supercharger.

$N/\sqrt{T_0}$ = 1,060
W = mass flow of air lb. per min.
N = r.p.m. of impeller
ρ_0 = initial air density at T ° C. abs.

friction loss and the general upset of the air-flow caused by the non-volatile liquid present on the impeller and diffuser vanes.

With certain assumptions as to the composition of the fuel and its latent and specific heats, it is possible to calculate the increase of the pressure ratio to be expected from complete evaporation of the fuel before entry to the impeller,[68] and the full-line curve of fig. 106 shows the calculated pressure ratio based upon the curve AA, observed with air only. The upper dotted curve through the observed points with petrol flowing shows that the actual rise was even slightly

greater than the calculated one for complete evaporation; but the difference between the dotted and the full-line curve would be eliminated by a change of 0·5 per cent. in the assumed specific heats of the air-fuel mixture at constant pressure and constant volume

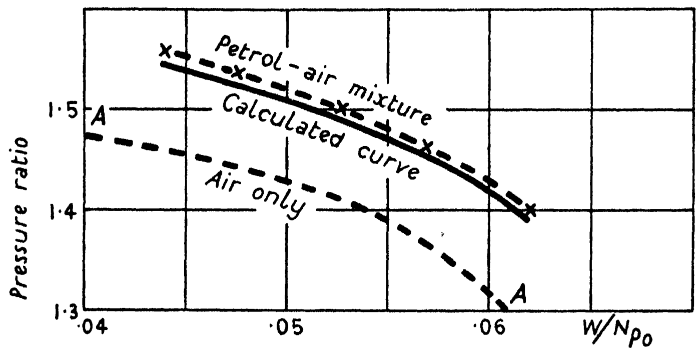

FIG. 106. Comparison of experimental results on petrol with the calculated effect of complete evaporation before entry to the impeller. Initial temp. 60° C. approx. 19,350 r.p.m.

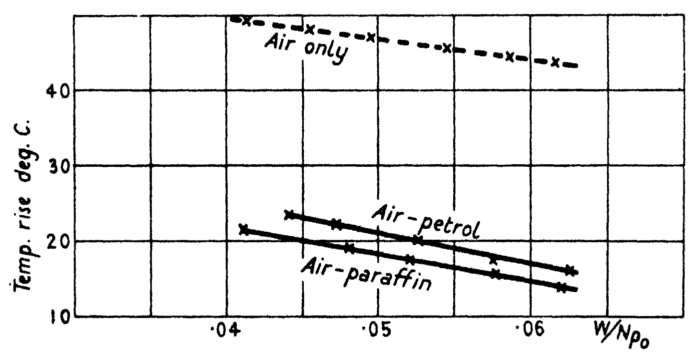

FIG. 107. Observed temperature rise in a supercharger of air and air-fuel mixtures.
$t = 61°$ C.
$N = 19{,}350$.

(positive in one case and negative in the other), so that from the results in fig. 106 one may conclude that, within the limits of accuracy possible, the calculation confirms that with air at 60° C. evaporation within the supercharger was complete. There is some confirmation of this, too, in fig. 105 which shows that at the high mass-flows, at any rate, there was scarcely any difference between the pressure ratios for $t_0 = 40°$ and $t_0 = 60°$ C.

The same point appears also in fig. 107 where the distances between the dotted and the two full-line curves show the lowering of

the temperature rise which occurred in the supercharger below that which was observed with air only. The lowering by the paraffin was actually slightly greater than by petrol in these experiments, at $t_0 = 60°$ C., a result which clearly shows that the evaporation of the paraffin and still more, therefore, that of the petrol, must have been complete. That the cooling was greater with the paraffin may have been due either to a difference between the latent heats or to some difference between the air-fuel ratios. With inlet-air temperatures of 40° and 20° C. the lowering of the delivery temperature by petrol was the same as when $t_0 = 60°$ C. within 1°, but with paraffin it was only 21° and 13° C. respectively. That the lowering of the delivery temperature by petrol was the same at all three inlet temperatures of 20°, 40°, and 60° C. is not inconsistent with the curves of fig. 105, which show a progressive increase of the pressure ratio, because the effect of the fuel upon the latter would depend upon how early during the compression process evaporation took place; whereas the lowering of the final temperature would depend merely on the total amount evaporated up to the point where the temperature was measured.

It should perhaps be made clear that in all these experiments the supercharger was enclosed within a chamber and surrounded with air at the temperature t_0. The rate of heat loss from the casing was therefore only affected indirectly by changes of the inlet air temperature between 20° and 60° C.

XI

THE TWO-STROKE CYCLE

ART. 63. *The importance of the method of fuel supply, and of a high thermal efficiency.*

As applied to large and comparatively slow-running engines, especially those of the marine type, the two-stroke cycle,* first proposed by Dugald Clerk in 1881, is, of course, widely employed. In the present chapter it is proposed to examine the possibility of developing a 2-cycle aero-engine which could compete with existing 4-cycle ones on a basis of weight-per-horse-power and fuel consumption; and the reasons why no such engine has so far appeared.

To the casual observer it might appear that to have twice the number of working strokes per revolution of the crankshaft would be a short cut to halving the weight per h.p. of the whole engine. How far this is so, however, will depend upon the possibility of maintaining sufficiently high M.E.P.s at corresponding speeds, combined with an equally good economy in fuel, and margin of reliability.

Comparing them first of all in general terms, one may regard the 2-cycle engine as one in which the pumping functions of the main pistons, which occupy every alternate revolution in the 4-cycle engine, have been replaced by one common pump for the whole engine—the scavenge blower—thus setting free the main pistons for twice the number of working strokes.

This points to a big advance for the 2-cycle engine in regard to weight per h.p.; but it is well to bear in mind that of two engines with cylinders of the same swept volume, the one a 2-cycle and the other a 4-cycle, the former would inevitably be the heavier. It must needs carry the scavenge blower, for one thing, and for the same length of stroke the pistons must be longer and heavier, and this in turn reacts upon the overall dimensions of the cylinders.

It may be objected that the majority of 4-cycle engines now carry some form of supercharger, and that the scavenge blower cannot therefore be fairly counted as an extra weight. To some extent this is true; but for several reasons the scavenge blower for a given size of engine would need to be larger and heavier than the corresponding supercharger of the normal type. Apart from its having to supply twice the swept volume at the same crankshaft speed, there is

* In what follows the two-stroke cycle will, for brevity, be referred to as the 2-cycle, and the normal engine as a 4-cycle engine.

necessarily some direct wastage of the scavenge air, and therefore of the power absorbed in compressing it. A further point is that the degree to which the scavenge blower could be used to maintain ground-level power at altitude is open to considerable doubt. There would certainly be no gain of power, at a constant speed and scavenge pressure, as there is in a 4-cycle engine by reason of the fall of pressure on the exhaust side. It is possible, therefore, that the 4-cycle engine without a supercharger should rather be the basis of comparison.

Fuel economy must also suffer, in the 2-cycle engine, from the need of driving a large scavenge blower; but this power loss can be offset against the extra friction and pumping loss during the idle strokes in a 4-cycle engine and, moreover, in some forms of the 2-cycle engine the thermal efficiency is helped by a lower heat loss to the cylinders than in a 4-cycle, as explained below in connexion with the 2-piston engine. A good fuel economy in the 2-cycle engine, like a high power output, depends primarily upon the efficient scavenging and recharging of the cylinder in the short time available for the process. If a high pressure is needed in the scavenging air, then the overall fuel economy will suffer through an excess of power being absorbed in driving the compressor.

In the 4-cycle engine the expulsion of the burnt gases and induction of the fresh charge occupy more than a whole revolution of the crank, about 460 deg., for the exhaust valve in a high-speed engine begins to open at least 50 deg. before the bottom dead centre, while the inlet valve does not close until 45 deg. or more after the same point on the next revolution. In the 2-cycle engine the whole series of processes of exhaust, scavenging out the burnt gases, and recharging the cylinder has to be concentrated within about 130 deg. of crank angle, from 75 deg. before to 55 deg. after the dead centre; and even this, it will be observed, cuts down the duration of the expansion, or working, stroke to no more than about 100 deg. of crank angle as compared with 130 deg. in the 4-cycle engine.

In this extreme shortness of the time available for recharging the cylinder lies the first, and fundamental, difficulty of applying the 2-cycle in a high-speed engine. The second drawback to the type as an aero-engine applies only to petrol engines in which the fuel and air are supplied together from a carburettor, as is normal in the 4-cycle engine.

It is inevitable that during a part of the short period available for exhaust-scavenge-recharge the inlet and exhaust ports should both be open together, and it is virtually impossible not to lose some of the air-petrol mixture supplied under pressure by the scavenge pump

before the exhaust valve has closed. The result is a wastage of fuel, and a fuel consumption per h.p. which would tell heavily against the type in the air, where extra fuel to be carried is equivalent to a heavier power-plant.

The difficulty of fuel wastage is eliminated in the Diesel type of engine using direct injection of a heavy oil fuel near the top of the compression stroke, and equally in an engine using spark ignition and a moderate compression ratio in which the fuel is injected into the cylinder and evaporated during the compression stroke.

This latter cycle of operations, in which the fuel is commonly a 'safety' fuel of higher flash-point than ordinary petrol, has been explored to some extent in single-cylinder research engines. It is doubtful, however, apart from the practical difficulties of starting and of a uniform distribution of the fuel, whether there is in principle much to be gained by employing the 2-cycle in a spark-ignited engine of moderate compression ratio. The problem of getting rid of the waste heat is the chief limiting factor upon which the power output per cylinder depends in the normal 4-cycle engine of to-day, and assuming the same thermal efficiency in the two types* this limit will operate at the same power output whether the engine employs the 2-cycle or the 4-cycle. In other words, the only advantage of employing a 2-cycle engine of moderate compression ratio would be that the present-day limit of power output per unit of swept volume, as set by the waste heat problem, would be reached at lower crank-shaft speeds, and reduction gearing to the air-screw would probably not be required.

If the need for the separate injection of the fuel into the cylinder of a 2-cycle engine be accepted, there seems every reason to pass at once to the 2-cycle Diesel engine with its compression ratio of 12 or 15 to 1, and greatly enhanced thermal efficiency. This is the form of 2-cycle engine which will be considered henceforward in this chapter, for it is in this form that one can say there are certain fundamental advantages that are worth striving for.

In the 4-cycle Diesel engine at the highest power output per cylinder reached to-day (in small cylinders) we are still far from the limit so far as waste heat is concerned, and this for two reasons:

(1) The heat generated per unit of swept volume is less, because it has not so far been found possible to burn, by means of an injected fuel-spray, more than about 75 per cent. of the contained air; and

* Owing to the limitation of the expansion stroke, already referred to, thermal efficiencies must tend to be lower, and the proportion of the waste heat higher, in the 2- than in the 4-cycle except, perhaps, in the 2-piston type of cylinder.

(2) Owing to the very high compression and high thermal efficiency of a Diesel engine the waste heat is a smaller proportion of the whole.

The result is that in the 4-cycle Diesel engine even at its maximum output there is no difficulty about keeping the exhaust valves and pistons reasonably cool; and the prospect, therefore, of doing so at whatever increased power output per cylinder may prove possible in a 2-cycle engine is reasonably good.

ART. 64. *Possible forms of the 2-cycle Diesel engine for aircraft.*

The problem of reaching a high power output per cylinder at high speeds centres in the first place upon the question of efficient scavenging. Immediately after the exhaust valve has opened, the cylinder, with the piston near the bottom dead centre, will be full of burnt gas at a temperature of 600°–800° C., and the incoming charge of fresh air has to sweep out these burnt gases and establish itself in their place during about 90 deg. of crank revolution (0·01 sec. at 1,500 r.p.m.). The pressures developed in the cylinder during the next cycle will depend directly on the completeness with which this change is made, because they will be in proportion to the amount of oxygen in the cylinder when the fuel charge is injected.

In several forms of slow-running engine the inlet and exhaust ports are disposed on opposite sides of the cylinder's periphery near the bottom of the stroke, and the path followed by the blast of scavenging air is then somewhat, but only roughly, as indicated by the dotted lines in fig. 108. The exhaust-valve ports are opened in advance of the inlet, so that the scavenge air follows upon the heels of the waste gases as they escape. An attempt is often made to prevent short-circuiting of the flow of the scavenging air direct from the inlet to the exhaust ports by shaping the top of the piston so as to deflect the air-stream from the inlet ports up towards the cylinder head.

The perfect scavenging process would be one in which the whole of the residual burnt gases were displaced by the incoming charge without any intermixing; but with a cylinder of the type shown in fig. 108 it is obvious that a great deal of general turbulence will be set up, and that the turbulence will produce a mixture of waste gas and fresh air. The higher the revolution speed, moreover, and the speed of entry of the scavenge air, the more complete will this mixing be, and the less efficient the scavenge for a given quantity of air supplied.

It has been found by experiment, and may be accepted as a *sine qua non*, that in any engine designed to meet aircraft requirements

some form of 'end-to-end' scavenge must be arranged in order to combine a high speed, say, 1,500–2,000 r.p.m., with an efficient sweeping out of the burnt gases and recharging with fresh air. Even with end-to-end scavenge it is a difficult problem unless one employs an inconveniently high pressure behind the blast of scavenge air. This pressure should not be more than 7 lb. per sq. in. if an exorbitant amount of the engine power, besides an excessive weight, is not

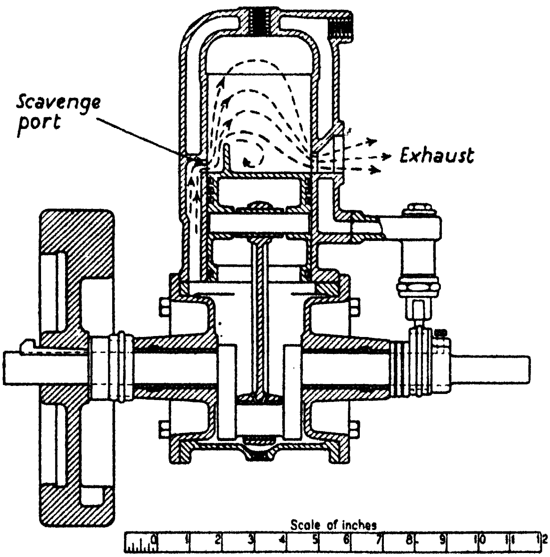

FIG. 108. Two-stroke cycle engine without valves.

to be expended on the scavenge pump. It must be remembered that whereas in a 4-cycle engine about half the work absorbed in driving a supercharger can be recovered directly as useful work on the pistons during the induction stroke, in the 2-cycle nothing is recovered, and moreover, for reasons which will appear later, the 2-cycle engine wastes from 25 to 30 per cent. of the air delivered and therefore requires a larger compressor for the same power output.

There appear to be three possible types of cylinder in which an end-to-end scavenge can be arranged. No one, at any rate, has so far suggested a fourth which looks attractive as a practical possibility.

1. *The two-piston engine.* In the first type, illustrated diagrammatically in fig. 109 and developed into a practical success in the Junkers' Jumo engine, each cylinder is provided with two pistons, each piston being connected, in the Junkers design, to its own crankshaft, one at each end of the cylinder. One piston controls the

inlet and the other the exhaust ports, there being no valve gear whatever. The motion of the two pistons is slightly out of phase, so that the exhaust ports at one end of the cylinder are opened by one piston some degrees before the uncovering of the inlet ports and the entry of the scavenging blast at the other end. Fuel injection takes place near the centre of the cylinder after the fresh charge has been compressed and heated by the approach of the two pistons one to another.

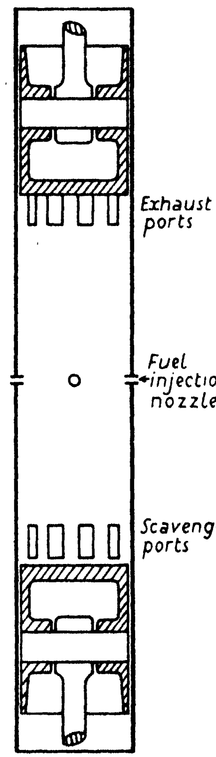

Fig. 109. Section of 2-piston cylinder with end-to-end scavenge.

A point to be noted about this type of cylinder is that the complete elimination of the cylinder head eliminates also that part of the waste heat which is normally lost to the head. It is the heat lost to the cylinder head, moreover, before the expansion has begun, that has an important effect upon thermal efficiency; so that the 2-piston type of cylinder starts with a fundamental advantage from the point of view of thermal efficiency, which has been reflected in the very low rates of fuel consumption per B.H.P. hour observed with the Jumo engine.

2. *Single-piston engine with poppet exhaust valve.* A second possible type of cylinder is that illustrated in fig. 110 in which there is only one piston per cylinder, and this controls the inlet ports exactly as one of the pistons in the Junkers design. The exhaust ports take the form of a poppet-valve in the cylinder head, operated by suitable external valve gear.

The alternative, in which exhaust ports in the cylinder barrel would be controlled by the piston and the scavenge air would enter through valves in the head, appears to have little to recommend it. Adequate cooling of the piston is likely in any 2-cycle design to be a critical factor, and this will be made doubly difficult where the piston controls the exhaust ports and is subject to scouring by the hot gases as the ports are uncovered. With a piston-controlled inlet, on the other hand, the piston is cooled by the scavenge air before any mixing with the hot cylinder contents has had time to occur.

3. *Single-piston engine with sleeve-valve.* A third type, which is a variant of the second with certain technical advantages, employs a sleeve-valve to assist in the control both of the inlet and exhaust.

The scavenge air enters, as before, through a ring of ports cut in the sleeve and in the cylinder liner, and opened up by the piston near the bottom of its stroke. The operation of the exhaust ports is independent of the piston. The valves depend for their opening and closing upon the registration of the ports cut in the sleeve with those cut in the cylinder liner, within which the sleeve works, as was illustrated in fig. 44 (p. 95). Besides that of not needing so long—and therefore so heavy—a piston, the sleeve-valve cylinder has an advantage over no. 2 above in that the design of the cylinder head is left free of all valves.

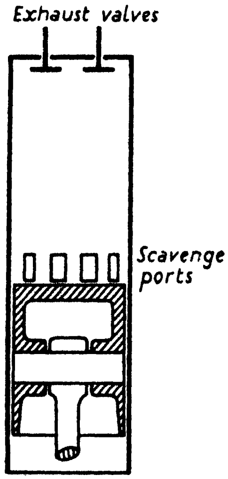

FIG. 110. Two-stroke cycle cylinder with poppet-type exhaust valves.

In this, as also in the other two types, the scavenge air can be given, if desired, an organized rotary motion about the axis of the cylinder. This is accomplished by the shaping and direction of the inlet ports round the periphery, through which the scavenge blast enters.

ART. 65. *The scavenging process.*

The extremely short time available in any 2-cycle engine for clearing out and recharging the cylinder has already been emphasized. In this article the features and some of the difficulties of the scavenging process will be examined in more detail, from the point of view of the interrelation between the valve-timing and the pressures in the cylinder if a satisfactory scavenge is to be obtained.

For the purpose of discussion it is convenient to regard the complete respiratory process of a 2-cycle engine as being made up of three distinct periods:

(1) the exhaust period, during which the exhaust valve only is open;
(2) the scavenging period, during which both inlet and exhaust valves are open together;
(3) the charging period, during which, if the exhaust valve is not actually closed, the inlet opening is at any rate considerably in excess of the exhaust, so that in either case the pressure in the cylinder builds up nearly to that of the scavenge air supply.

As indicated already in art. 62, the total period of time available for all three stages is limited to that corresponding to about 130 deg. of crank angle, from 75 deg. before the dead centre to 55 deg. after it.

In all but the sleeve-valve cylinder the inlet ports would be entirely piston-controlled, and it follows that their opening and closing must be symmetrical about the bottom dead centre. It is essential that they should close not more than about 50 deg. after the dead centre, otherwise too much of the compression stroke would be wasted, and this limits the total period of opening for the inlet port to about 100 deg. of crank angle.

The adjustment of gas velocities and port areas such as would normally be made for a large slow-running engine becomes somewhat superfluous at the speeds required in the aeroplane field, for the fact is that every possible square inch of port area will be required: in the exhaust ports so as to allow the pressure to drop quickly, after they begin to open, to below the scavenge pressure; and in the inlets so as to keep down the necessary scavenge pressure to a minimum, and so also the work done in producing it.

The design problems in the high-speed engine from the scavenging point of view are those of utilizing all possible port area, and of dividing up the time available for each of the three periods to the best advantage. For period (1), just sufficient time×area-of-opening must be provided to allow the pressure of the exhaust gas to fall to that of the scavenge air before period (2) begins. If it does not, some blow-back of the hot exhaust gas through the inlet ports will occur as they open, and its subsequent return, along with the scavenging air, will spoil the scavenge. On the other hand, if period (1) is too long it encroaches needlessly on the time required for the other two stages. During period (2) the simultaneous opening of both sets of ports must give time for a large proportion of the residual exhaust gas to be scoured out by the inrush of scavenging air; while during period (3) the time×area product for the scavenge-port opening must allow of the pressure in the cylinder being built up as nearly as possible to that of the scavenge air supply.

When the maximum areas available for exhaust and inlet ports are about equal, then a lead of about 35 deg. is necessary for the exhaust valve, in a single-piston cylinder, in order to allow the gases to escape and the pressure in the cylinder to fall adequately before the scavenge ports open. In the 2-piston engine, where both scavenge and exhaust ports are piston-controlled, the lead of the exhaust port opening is obtained very simply by adjustment of the length of the ports, combined with a difference of phase between the cranks at the exhaust and inlet ends of the cylinder. In the Junkers' Jumo engine the exhaust crankshaft has a lead of about 9 deg. over the other. The length of the exhaust ports is such that they begin to open 75¾ deg. before the dead centre of the exhaust

crankshaft; and the inlet ports open about 18 deg. later. The exhaust ports must, of course, close $75\frac{3}{4}$ deg. after their own piston's dead centre, and the lag of the scavenging piston allows the scavenge ports to close at the same moment.

If the exhaust ports were controlled by a poppet-valve, actuated by a cam (as in type 2 of the last article), then the rate of opening and closing of this, when excessive accelerations are avoided, would have to be a good deal slower than that of piston-controlled ports. It would, in consequence, be very much more difficult to provide an adequate time × area product within the crank angle available, if it were not that the charging of the cylinder nearly up to the scavenge air-supply pressure can be effected, with the more slowly closing exhaust valve, without actually completing the closure before the inlet. If the two are finally closed at the same moment there will have been, during the last 30 deg. or so, a large excess of inlet over exhaust area, which allows the pressure in the cylinder to build up.

The same arrangement is possible in the sleeve-valve cylinder, where the exhaust port opening and closing is controlled entirely by the sleeve. Fig. 111 shows the relationship between the exhaust and inlet port opening diagrams which has been found by Ricardo to fulfil the required conditions for a sleeve-valve cylinder of bore and stroke $5\frac{1}{2}$ in. × 7 in. up to speeds of 1,300 r.p.m. The exhaust port opening extends over a total period of 130 deg., while the inlet closes at the same moment after an opening of 102 deg. The inlet opening is controlled primarily by the piston, but it can be delayed and its phase altered by alteration of the sleeve motion. It will be observed that instead of a port opening symmetrical with respect to the dead centre, that shown in fig. 111 extends from 47 deg. before to 55 deg. after it.

The area A denotes the time integral of the exhaust port area during period (1), up to the end of which there is only a negligible opening of the inlet, or scavenge port. From then onwards, until 55 deg. after the dead centre, which covers periods (2) and (3), both sets of ports are open together, but from about 10 deg. after the dead centre there is a large excess of inlet over exhaust port area, marked C on the diagram, and this enables the pressure in the cylinder to build up to well above atmospheric before the simultaneous closure of both valves. The sequence of events during the critical period is clearly shown in the light spring indicator diagram in fig. 112. This was taken while the engine was running at 1,200 r.p.m. and giving a B.M.E.P. of 98·5 lb. per sq. in., with a maximum cylinder pressure of 800 lb. per sq. in.

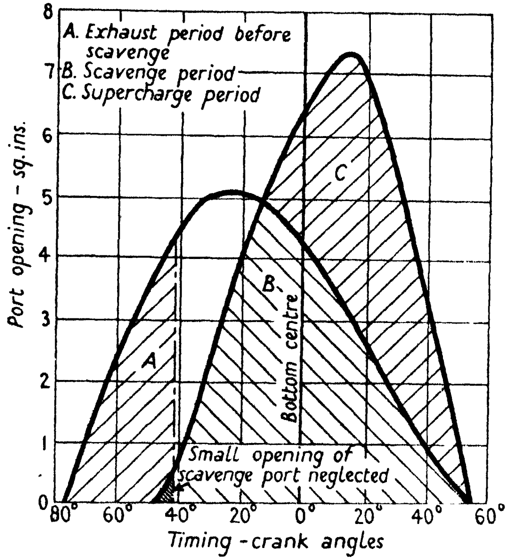

FIG. 111. Valve-opening diagram, 2-cycle sleeve-valve cylinder.

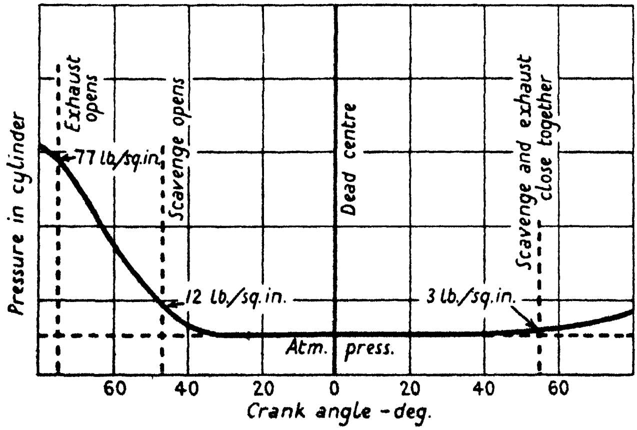

FIG. 112. Indicator diagram, 2-cycle engine from 80 deg. before to 80 deg. after the dead centre. Sleeve-valve cylinder $5\frac{1}{2}$ in. $\times$ 7 in. Speed 1,200 r.p.m. B.M.E.P. 98·5 lb. per sq. in. Scavenge pressure 4·8 lb. per sq. in.

The exhaust port began to open at 75 deg. before the dead centre, when the pressure in the cylinder was 77 lb. per sq. in., and after this the pressure fell rapidly to 12 lb. per sq. in. at the point, 47 deg. before the dead centre, when the scavenge port began to open. At 40 deg., by which time the inlet opening had become appreciable, the cylinder pressure had dropped to just about the scavenge pressure of 5 lb. per sq. in., and very soon after that it became atmospheric and remained there until about 20 deg. before the closure of the valves. During the last 20 deg. the pressure rose gradually to 3 lb. per sq. in., and thereafter more rapidly during the compression stroke with all valves closed.

In any type of cylinder affording an end-to-end scavenge the ratio of the cylinder diameter to the length between the scavenge and exhaust ports is an important feature which affects the efficiency of the scavenging process. Between the 2-piston and the single piston types, whether the latter be of type (2) or (3) (see p. 289), there is an important difference in this respect, for in the 2-piston engine the ratio of the length between the ports to the cylinder diameter will be about twice what it is in the other type. To explain the significance of this, reference must be made to some work by the late Prof. Hopkinson.[54]

In these experiments the flow conditions during the scavenging of cylinders with a set of peripheral ports at each end were studied by means of observations on the flow of water. The experiments were based upon the fact that the character of the flow in geometrically similar cylinders and valve passages will be identical, whatever the fluid, provided the Reynolds Number Vl/ν is the same. The experimental cylinder was full size in Hopkinson's experiments, and since the kinetic viscosity, ν, for air is thirteen times that of water, the length of the scavenge period and the flow velocity through the ports can be altered in the same ratio to a convenient order of magnitude, when applied to an engine of 250 r.p.m., for the scavenge period becomes rather more than 1 second. Although the quantitative application of these experiments is to a slow-running engine, there can be little doubt that the influence of the length/diameter ratio upon scavenging efficiency which they demonstrated would apply also at higher speeds.

Hopkinson pointed out that if it be assumed that instantaneous and perfect mixing of the scavenge air with the cylinder contents occurs upon entry, then the volume of air, y, which is retained in the cylinder when the exhaust valve closes, is related to the volume x which has entered through the inlet ports, by the equation

$$y = 1 - e^{-x},$$

y and x being expressed in terms of cylinder volumes or, to a sufficient approximation, in terms of the swept volume of the piston or pistons.

This theoretical relationship between the air supplied and the air retained in the cylinder is shown graphically in fig. 113 by the lower full-line curve. This curve and the experimental curves which lie

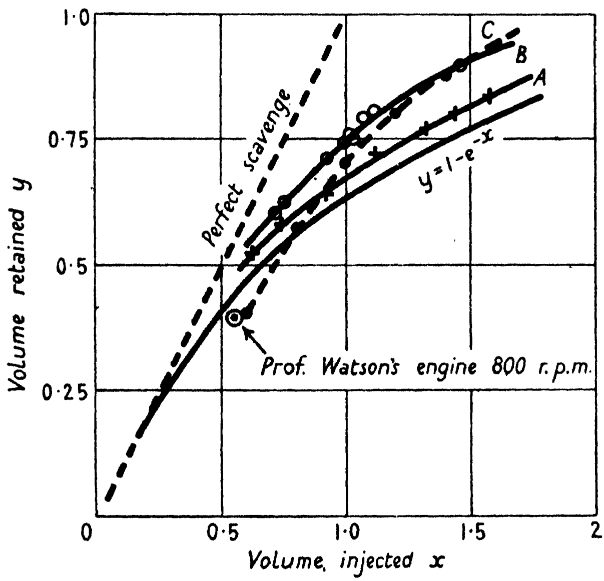

FIG. 113. Relation between volume injected and volume retained for a cylinder with end-to-end scavenge, showing theoretical curve for perfect mixing and experimental results from 3 different cylinders.

Curve *A*. Hopkinson and Bird. Cylinder 8 in. × 10 in.
" *B*. Hopkinson and Bird. Cylinder 8 in. × 20 in.
" *C*. Deduced from Ricardo. Cylinder $5\frac{1}{2}$ in. × 7 in.

above it apply only to events which take place at constant pressure, but they are none the less valuable for the way they bring out some of the essential characteristics of the scavenging process. The theoretical relation between the fluid injected and fluid retained is well confirmed by the experimental results obtained with water, which are due to A. L. Bird[55] and are shown in the curves *A* and *B* of fig. 113, the one being for a cylinder 8 in. diam. and 10 in. long, and the other for a cylinder of the same diameter but twice as long.

The theoretical curve assumes perfect mixing of the scavenge air with the exhaust gas, but that is not, of course, the condition to be aimed at. In the ideal scavenging process there would be no mixing at all. The advancing front of the scavenge air would drive out all the

exhaust gas before it without the loss of any of the air through the exhaust valve, so that after this had closed the relation between the volume injected and the volume retained would be that shown by the dotted straight line, namely equality.

It is due to the memory of Dugald Clerk to record that he evidently understood perfectly what was required in order to achieve the best possible scavenge; and that the scavenging in his earliest 2-cycle gas-engine of 1881 was probably superior to anything achieved to-day. The reason for its excellence was that, in a horizontal engine, Clerk was not limited as to the length of his cylinder, and he employed a long conical combustion space and end-to-end scavenge. The scavenging blast was able to lose velocity as it expanded down the conical end of the cylinder and thereafter to drive out the residual gas before it, with a minimum of intermixing.

The theoretical curve does not express the worst possible case, for an even worse one would be that in which there was a direct passage of the scavenge air from the inlet to the exhaust ports, without mixing with the exhaust gas. This does in fact occur to a considerable extent in the simple valveless 2-cycle engine without end-to-end scavenge, illustrated in fig. 108, in which the scavenge and exhaust ports are opposite to one another at one end of the cylinder. Conditions represented by the point lying below the theoretical curve in fig. 113 were obtained during tests of such an engine by Watson.[56] To minimize the amount of air wasted directly to the exhaust ports, engines with side-to-side scavenge have the air inlet passages designed so as to direct the flow towards the cylinder-head, and the top of the piston is often specially shaped for the same purpose.

The flow of air during the scavenging of a cylinder of this type has been examined experimentally by Lindner.[73] Two quite different conditions of flow were observed, according to the amount of the port openings. At first a large vortex filling nearly all the space above the piston was set up, but as the piston uncovered more of the ports there was a sudden change of configuration, showing the lower part of the cylinder occupied by direct flow lines between the inlet and exhaust ports, the rest of the space being filled by a new vortex of opposite rotation to the first one. Since it may be expected that the cores of the vortices consist mainly of the residual products of combustion, which thus evade being scavenged, it seems likely that the formation of vortices is a fundamental difficulty in the way of securing an efficient scavenge in this type of cylinder.

The chief practical interest of Hopkinson and Bird's work lies in the demonstration that a substantially more efficient scavenge is obtained when the distance between the ports is doubled, as it is in

the 2-piston as compared with the single-piston type. It appears from fig. 113 that if a fresh charge equal to 0·8 of the cylinder volume is to be retained at the moment when the exhaust valve closes, this would only require a volume of scavenge air equal to 1·1 × the swept volume in the long cylinder, but 1·45 × the same volume in the short one. To give the same mixture during compression, therefore, in two engines of the same bore and stroke, the one having

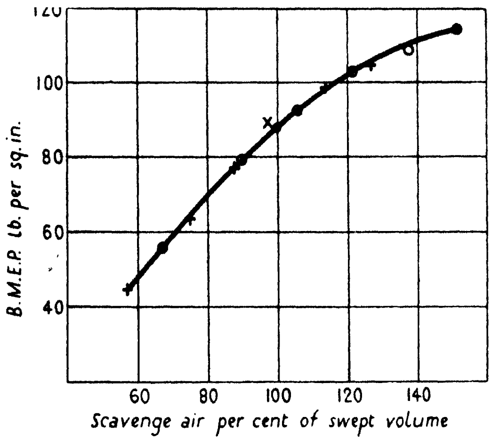

FIG. 114. Relation of B.M.E.P. to scavenge air supply measured at room temperature and pressure. Sleeve-valve 2-cycle cylinder, 5½ in. × 7 in., at 800 r.p.m.

Observations with two different valve port arrangements indicated thus +, and thus o.

6 cylinders of 2 pistons each and the other 12 cylinders of half the length, the air to be supplied would be in the ratio 1·1 to 1·45 in favour of the 2-piston type. In that type, however, the air would have to be supplied through 6 sets of ports only, instead of 12, so that in spite of the more efficient scavenge there may be no saving of the power absorbed by the compressor unless a larger inlet port area per cylinder can be contrived in the 2-piston type of engine; and there is no obvious reason why that should be so.

It happens that the ratio length/diameter in the shorter of the two experimental cylinders—in Bird's experiments—was almost exactly that in Ricardo's sleeve-valve cylinder, of bore and stroke 5½ in. × 7 in., and an interesting light upon the scavenging efficiency of that cylinder, for comparison with theory and with Hopkinson and Bird's experiments can be derived from the observations of the power output obtained under different conditions.

Fig. 114 shows how the B.M.E.P. was observed to increase as

the amount of scavenge air supplied per revolution was increased by raising the scavenge pressure. In an engine of the same-sized cylinder working on the 4-stroke cycle, but otherwise identical, it was found that a B.M.E.P. of 125 lb. per sq. in. was obtained when normally aspirated and running at the same speed of 800 r.p.m. It will be seen that the B.M.E.P. in the 2-cycle engine was 88 lb. per sq. in. when the air supplied was 100 per cent. of the swept volume (77 cu. ft. per min. at 800 r.p.m.). If we may assume, as a rough approximation,

(1) that the engines were of the same mechanical efficiency;
(2) that scavenging was perfect in the 4-cycle engine, so that the air retained was equal to the swept volume;
(3) that the combustion efficiency for the air was the same in each,* so that the observed B.M.E.P.s in the 2-cycle engine were proportional to the quantity of air retained under each of the different conditions of scavenge,

then it follows that at the point X, for example, in fig. 114, corresponding to $x = 1{\cdot}0$ in fig. 113, the ratio of the volume of air retained to the volume supplied, each measured under atmospheric conditions, was

$$\frac{88}{125} = 0{\cdot}7.$$

The relationship between the air retained and the air supplied at other scavenge pressures in Ricardo's experiments may be deduced in the same way from the curve of B.M.E.P. against air supply in fig. 114, and the results are shown by the curve C in fig. 113.

Art. 66. *Power output and air consumption.*

The available data from tests of 2-cycle engines of the high-speed type is too meagre to allow of drawing very general conclusions as to the relation between power output and air supply, and the best plan appears to be to give the results of experiments, such as they are, on different types of engine, so far as possible in a comparable form.

In Watson's tests[56] the engine was of the simple valveless type illustrated in fig. 108 with crankcase compression and without end-to-end scavenge. High mean pressures were therefore not to be expected, and since the air supply to the cylinder depended upon the

* For some reason not fully explained the combustion efficiency for the fuel in the 2-cycle sleeve-valve engine, as judged by the minimum fuel consumption per B.H.P. hour, has never been so high as in the 4-cycle. It does not follow, however, that at full load the fraction of the air present which was burnt was not the same in each.

amount drawn into the crankcase during the compression stroke, it was to be expected that wire-drawing at the crankcase inlet port would cause a fall of M.E.P. as the rotation rate was increased. The effect could be minimized by a wise choice of the port timing, as can be clearly seen from the two curves of total air supply in fig. 115. These show the total air per cycle as a fraction of the swept volume. The full-line curve corresponds to an opening of the inlet port to the

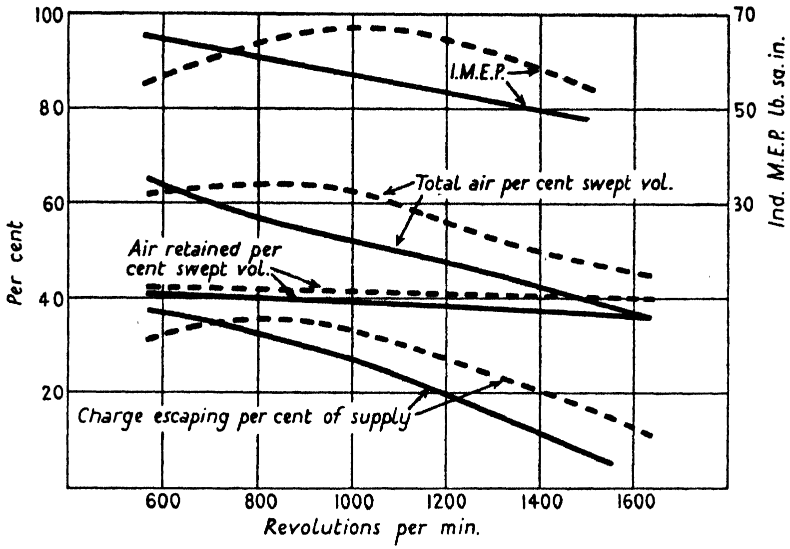

FIG. 115. Curves of total air, air retained, and I.M.E.P. in a valveless 2-cycle engine with crankcase compression.

crankcase for $\pm 34\frac{1}{2}$ deg. and the dotted curve for ± 41 deg. on each side of the top dead centre. The longer opening allowed the maximum air charge to be maintained up to 900 r.p.m.

In this engine all the port openings must be symmetrical about the dead centres, since they are piston-controlled; and if a lead of the exhaust port opening is arranged, to allow the exhaust gas to get away, that means that the port cannot close until after the inlet to the cylinder. In Watson's engine the exhaust port was open for ± 61 deg. and the inlet to the cylinder for $\pm 48\frac{1}{2}$ deg. on either side of the bottom dead centre.

The curves of air retained, in fig. 115, show the quantities of air taking part in the combustion, at different speeds, as estimated from analyses of the gases in the exhaust pipe. It will be noticed that there is only a very small drop with increase of speed, which indicates that the wire-drawing at the inlet ports to the crankcase was compensated

to a large extent by a diminution of the air directly lost through the exhaust valve before the inlet port to the cylinder closed. Furthermore, the increased port opening to the crankcase, as shown by the dotted curve, produced only a trifling rise of the air retained per cycle. This is easy to understand, for the exhaust port closed 12½ deg. after the inlet, and there is no possibility, therefore, of the pressure before compression being other than atmospheric. The curves of fig. 115 may be interpreted as meaning that the extra air supply derived from the altered port timing went almost entirely to improving the scavenge. The small increase in the amount of air retained was due to the lower mean temperature of the cylinder contents owing to the greater dilution of the residual gas by the scavenge air. The maintenance of the amount of air retained, as the speed was increased, in spite of the fall in the total amount of air inspired, is to be explained by the fact that as the crankcase compression pressure became less at the higher speeds owing to the wire-drawing, there was less scavenge air short-circuited directly from the inlet to the exhaust ports.

The thermal efficiency of Watson's engine, as estimated from indicator diagrams, was exceedingly good, and the M.E.P.s shown in fig. 115 would probably be difficult to improve upon very much in a valveless engine with no end-to-end scavenge, although an increase of 10 or 15 per cent. could no doubt be made by using a higher compression ratio, for that was only 3·92 in Watson's engine.

A figure of at least 90 lb. per sq. in. must be aimed at as the B.M.E.P. in a 2-cycle aero-engine, before allowance is made for the power to drive the scavenge blower, and it is clear, therefore, that the simple valveless engine does not come within sight of competing with the modern 4-cycle petrol engine on a weight per h.p. basis.

It is noteworthy that in spite of the constancy, over the speed range, of the air retained in the cylinder and taking part in the combustion, the M.E.P. in Watson's engine varied as shown by the two upper curves in fig. 115. As compared with the ratio 0·95 between the air retained at 1,500 and at 1,000 r.p.m., the ratio of the M.E.P.s was 0·85. So long as the thermal efficiency of an engine can be maintained, the indicated power output must go hand-in-hand with the weight of air entering into combustion, and the deduction to be drawn from Watson's experiments is probably that with so large a fraction as 60 per cent. of burnt gas always in the cylinder the rate of combustion was too slow for the maintenance of the efficiency at high speeds. This is borne out by an examination of the indicator diagrams which are reproduced in the original paper.

In Ricardo's engine, working on the Diesel cycle with injected fuel, no direct evidence could be obtained from exhaust gas analyses as to the proportions of the scavenge air which escaped and were retained. An indirect estimate was made in the last article by comparison with a similar 4-cycle engine, and the conclusion reached was that when the scavenge air was 100 per cent. of the swept volume, 70 per cent. of it was retained in the cylinder.

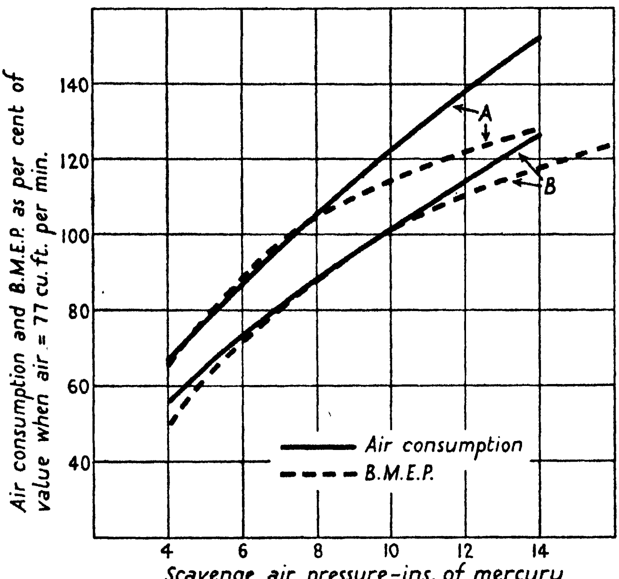

FIG. 116. Increase of air consumption and B.M.E.P. with scavenge air pressure in a sleeve-valve cylinder for two different arrangements, *A* and *B*, of port areas and timing. Speed 800 r.p.m. B.M.E.P. 88 lb. per sq. in. with either arrangement when the air consumption was 77 cu. ft. per min. (100 per cent. of swept volume).

Fig. 116 shows how the air consumption and the B.M.E.P. in Ricardo's cylinder varied with the scavenge pressure for two different arrangements of port area and port timing, distinguished as conditions *A* and *B*. The air consumption and B.M.E.P. are each shown as a percentage of their values when the air taken was equal to the swept volume. At this value of the air consumption the B.M.E.P. was almost identical with either arrangement *A* or *B*, at 88 lb. per sq. in.; but reference to fig. 116 shows that it required a scavenge pressure of 9·8 in. of mercury with arrangement *B* and only 7·4 in. with arrangement *A* to get this amount of scavenge air in through the inlet ports. It has already been mentioned (see fig. 114) that the B.M.E.P. was the same at the same flow of scavenge air even with wide variations of port area and port timing,

but the relationship illustrated in fig. 114 did not bring out the improvement shown by state A over state B in regard to the pressure needed to produce a certain flow of air through the cylinder.

The curves in fig. 116 suggest that the air retained in the cylinder

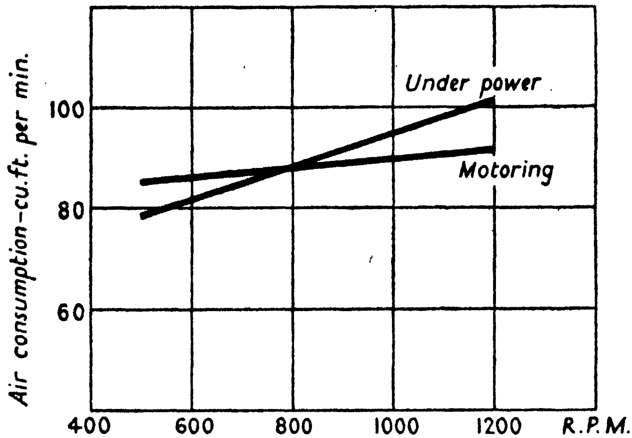

FIG. 117. Scavenge air-flow at various speeds when supplied at a constant pressure of 12 in. of mercury. Sleeve-valve cylinder 5½ in. × 7 in. Port timing as in fig. 111.

and burnt, to which the B.M.E.P. would be proportional, was approximately a constant fraction of the total air supplied, up to the point when the latter equalled the swept volume; but that with increasing scavenge pressures there was a greater and greater

TABLE 43

Variation of air consumption with speed in Ricardo sleeve-valve cylinder 5½ in. × 7 in. when being motored and when running under power. Scavenge pressure constant at 12 in. mercury. Port timing as in fig. 111.

	Total air taken by the engine	
	cu. ft. per min	*cu. ft. per rev.*
When motored	$80{\cdot}7+0{\cdot}0086N$	$0{\cdot}0086+\frac{80{\cdot}7}{N}$
When under power . . .	$62{\cdot}5+0{\cdot}033N$	$0{\cdot}033+\frac{62{\cdot}5}{N}$

loss of air during the scavenging process, as shown by the falling away of the curves of B.M.E.P. compared with those for the total amount of scavenge air supplied.

The effect of engine speed upon the air taken at a constant scavenge pressure is interesting, for the linear relation between the

two, illustrated in fig. 117 for a scavenge pressure of 12 in. of mercury, shows that the air passing into the engine per stroke can be regarded as made up of two parts, the one constant and the other inversely proportional to the speed. The actual volumes of air in cu. ft. per min. and per rev. measured at room temperature and pressure are given in table 43. The swept volume of the cylinder, V_s, was 0·0963 cu. ft. per rev. Fig. 117 and table 43 refer to results with arrangement *B* of port timing and area.

In this engine there was no definite demarcation between the scavenge period (2) and the charging period (3), as was explained in the last article, so that the interpretation of the figures of table 43 can be little more than qualitative. At the same time the existence of a term which is proportional to $1/N$ to represent the air-flow before there was any restraint at the exhaust valve, accompanied by a constant term to represent the charging up of the cylinder, is what would be expected if the two periods were quite distinct. And furthermore, the fact that the variable term proportional to $1/N$ should not differ widely between the conditions of being motored and of running under power, whereas the constant term was increased about fourfold when the engine was running under power, was again what might be expected; for in these circumstances the residual gas must have been at about 1,100° C. abs. instead of at atmospheric temperature, as it would be when being motored, and the cooling of the residual gas by the scavenging air would lead to the admission of more fresh air during the charging period.

TABLE 44

Air taken at 800 *and* 1,300 *r.p.m., when being motored and when under power, subdivided between the apparent scavenging and charging periods.*
Scavenging pressure constant at 12 *in. mercury.*

		Air taken in cu. ft. per rev.	
	Speed r.p.m.	*scavenging period*	*charging period*
When being motored . .	800	$1{\cdot}05V_s$	$0{\cdot}09V_s$
When under power . .	800	0·80 ,,	0·34 ,,
When being motored . .	1,300	0·64 ,,	0·09 ,,
When under power . .	1,300	0·50 ,,	0·34 ,,

The figures for the air consumption per revolution at 800 and 1,300 r.p.m., deduced from fig. 117, are given in table 44 in terms of the swept volume of the cylinder. At 800 r.p.m., as it happens, the air taken by the engine was the same whether motoring or

running under power, namely 0·1096 ($= 1{\cdot}14V_s$) cu. ft. per rev., but it will be seen that the subdivision is very different. The figures in the table suggest that for a certain scavenge pressure the air retained in the cylinder when working under power may be regarded as made up (*a*) of a quantity associated with the charging period and independent of the speed, equal to $0{\cdot}34V_s$, and (*b*) of a quantity associated with the scavenging period which decreased with the increase of speed, from $0{\cdot}80V_s$ to $0{\cdot}50V_s$ between 800 and 1,300 r.p.m.

ART. 67. *Piston problems of the 2-cycle engine.*

The simplicity of piston-controlled ports without any valve gear being one of the virtues of a 2-cycle engine, we may assume that the inlet ports would always be operated for preference in this way; more especially as the scavenge air with this arrangement helps to cool the piston, and because any mechanically operated valve-gear presents special difficulties for the 2-cycle engine at high speeds.

A little consideration will show that when a ring of ports are piston-controlled the length of the piston must be equal to, or in practice rather greater than, the length of the stroke. Otherwise the ports will be uncovered by the lower end of the piston-skirt when at the top dead centre, and air from the scavenge pump will be free to pass directly into the engine crankcase. In the 2-piston engine, with piston-controlled exhaust ports, the same considerations apply at the exhaust end of the cylinder, for although the gases in the exhaust pipe will not continuously be under pressure, like the scavenge air, nevertheless with a common exhaust pipe to several cylinders there would be trouble from intermittent blow-back of exhaust gas into the crankcase.

That one should be faced in this way with so large a minimum length of piston is a serious point for the 2-cycle aero-engine. Not only will the pistons be heavy and the piston friction large, the latter depending largely, as it does, upon the area of the oil film between the piston and cylinder, but also, and no less important, there is the influence of the long piston upon the distance of the cylinder head from the crankshaft centre line. As affecting the piston friction it should be added that since the piston has at all times to seal the ports there are only very limited possibilities of reducing the diameter over certain parts of it so as to reduce the area of contact and the area of the oil film to be sheared.

A further point of difficulty is that unless the piston carries an effective scraper ring at the bottom edge of its skirt, the ports themselves will tend to act as scraper rings, so that on the down-stroke oil

will collect on the edges of them, ready to be blown into the cylinder by the scavenge air, and into the exhaust pipe also in a 2-piston engine. This scraper ring at the bottom of the piston skirt must at all times remain below the ports: a condition which means not only that the piston must be longer than the stroke, but that the cylinder liner must extend for a whole stroke length below the ports. Otherwise the scraper ring would come out of the cylinder.

All these conditions are inherent in the 2-cycle engine with piston-controlled ports, and they all make for long and heavy cylinders without the extra power given by a long stroke. The stroke is seriously limited by the obliquity of the connecting-rods, which must not foul the inner ends of the cylinder liners. In the 2-piston engine the weight of the long cylinders is compensated by the absence of cylinder heads; but, on the other hand, there must be two crankcases, two crankshafts, and a train of gears to connect them. The result of these general considerations tends to limit the types of engine with a good prospect of success to the 2-piston type, and the single piston with a sleeve-valve. With this latter type of valve the need for a scraper ring on the piston skirt still holds good, but by employing the sleeve to assist in controlling the ports the length of the piston can be reduced by the length of the sleeve stroke. This would be about 20 per cent. of the main stroke. As compared with a piston of length about 105 per cent. of the main stroke, therefore, it can be reduced to about 85 per cent., with a corresponding saving in the overall size of the cylinder and in engine weight.

With any piston-controlled port it is nominally the top edge of the piston which controls the port timing. Actually, however, since the clearance of the top land on the piston cannot be cut very fine because of allowing for expansion, it may often be the top piston ring which is the effective seal on the port. The deeper the top land, the less positive will be the opening and closing of the port, and the more blow past the land there will be. On the other hand, a shallow top land exposes the piston-ring in its groove to more severe temperature conditions and danger of gumming up. If the ring becomes stuck it can no longer produce an effective seal, the port timing and engine performance are affected, and seizure of the piston will not be long delayed.

These considerations apply more acutely in the 2-piston engine of the Junkers type in which one of the pistons has to seal an exhaust port, for then it is burnt gas at over 1,000° C. which leaks down past the top land to the first ring as soon as this has passed the top of the exhaust port. In the sleeve-valve engine, where the piston controls only the inlet ports, the scouring of the top land by hot exhaust gas

is absent, and the danger of ring gumming is much less. If the top ring should become overheated, however, it is subject to conditions nearly as bad for the gumming of the oil, for although the scavenging air is comparatively cool it is richer in oxygen than the exhaust gases.

From the foregoing considerations it will be clear that the life of a 2-cycle piston under conditions of high speeds and high M.E.P.s will depend upon skilful design, and upon the closest possible attention to the problem of devising the most effective means of keeping it cool. If one compares two cylinders, a 2-cycle and a 4-cycle, of the same size and giving the same power output, with the same maximum pressures, then the shorter expansion stroke of the 2-cycle must mean that the proportion of waste heat will be higher and the cooling of the piston more difficult. The whole object of the 2-cycle being to give a greater power output in unit time, one has to face the fact that the amount of heat passing to the piston per sec. will be proportionately greater than in the 4-cycle engine of the same size of cylinder. On the other hand, the area of contact of the skirt with the cylinder wall will be nearly twice as great.

This is not to say that the means for getting rid of the heat will be twice as effective, for reference to Chapter IV will show that a piston depends very largely on the rings and ring-belt for its cooling, and that in most designs the skirt is comparatively unimportant. In a 2-cycle piston it will probably pay to design it so that all possible use is made of the large skirt for cooling purposes.

A point which adds to the severity of the conditions in regard to the gumming of the top piston ring is the fact that it is continually held, either by inertia or gas pressure, against the lower side of its groove. In the 4-cycle engine the top ring, like all the others, will be carried by inertia, at the end of the exhaust stroke, up against the upper side of the groove. With the probable lower speed of the 2-cycle engine it is likely that the gas pressure at every revolution prevents this, and the consequence is that there is no disturbance of any gummy oil once it has formed in the groove and nothing to prevent its steady progress towards the congealed cement which leads to a stuck ring.

The absence of force reversal at the gudgeon-pin bearing, likewise, makes the lubrication of this a difficult problem, and designs which have been found satisfactory in the 4-cycle engine will not be found adequate for the special conditions of the 2-cycle. Ricardo has developed an ingenious design of piston-connecting rod bearing of spherical form. The force between the two being always a thrust it is only necessary to hold the spherical end of the connecting rod lightly in the piston bearing, and the piston is free to rotate about

the axis of the cylinder if it wishes. This extra degree of freedom has proved invaluable in assisting to keep the rings free and in preventing local heating of the piston. In the sleeve-valve type of cylinder the movement of the sleeve encourages rotation of the piston, and this assists lubrication both of the spherical bearing at the small end of the connecting rod and at the periphery of the piston. A further point in favour of the spherical ball form of joint is the convenience with which it lends itself to an arrangement for cooling the piston by oil circulation.

ART. 68. *Comparison of 2-cycle and 4-cycle.*

A proper conclusion to this chapter would be an estimate of the future prospects and possibilities of the 2-cycle Diesel engine for aircraft, and some comparison with the 4-cycle, either of the Diesel or of the normal petrol-burning type. Any forecast, however, about future types of engine must be very much in the air. The 2-cycle has certainly not yet appeared in its final form, so that it is difficult to find a firm basis for comparing it even with the 4-cycle Diesel engine, and still more for comparing it with what the petrol engine may be in 5 years' time.

Apart from the governing questions of weight per h.p. and fuel economy there are certain mechanical points to be mentioned in favour of the 2-cycle engine. They are secondary, but may prove of great importance in the matter of reliability if the engine can justify itself on other grounds.

One of the serious difficulties of the Diesel, or compression-ignition, cycle as compared with the petrol engine is the wide torque fluctuation in relation to the mean torque, and the fact that high maximum pressures are experienced in the cylinders even at light loads. For a given number of cylinders the 2-cycle engine scores heavily in its greater uniformity of turning moment, at any rate in the radial form of engine. For the V-type with the cylinders in line it may be difficult to reconcile the requirements of dynamic balance with those of uniform turning moment. If it should be found necessary to fire two cylinders simultaneously in order to secure dynamic balance, then the advantage in regard to the even turning moment would be destroyed. This would be so, unfortunately, in a 12-cylinder engine with the two banks of cylinders set at 60 deg., although the difficulty can be got over by setting the cylinders at the less convenient angles of 45 deg. or 90 deg.

In a 2-cycle radial engine there is the further advantage that the inertia forces, all concentrated at the single crankpin, would be relieved at every stroke by the gas forces. Inertia loads on the crankpin are a

limiting factor in the radial engine at high speeds, and although the amount of the advantage cannot be assessed except in terms of a detailed design, there is little doubt that from the point of view of the mechanics of the engine there should be a substantial advantage in 2-cycle operation.

A point which is not so favourable in the 2-cycle engine is the steadiness of the load between the pistons and connecting rods. It is true this should make for smoother and quieter running, even when ample clearances are allowed at the big-end and gudgeon-pin bearings, but, as mentioned in the last article, the absence of force reversals in the bearings, more especially in that of the gudgeon pin which oscillates only over a small arc, must certainly hinder their lubrication. During long-sustained runs on experimental 2-cycle engines Ricardo has found this matter of the gudgeon-pin bearing to be a frequent source of trouble. The bearing surfaces, both on the connecting rod and on the piston, have shown a tendency to 'pick-up' which has proved extremely difficult to deal with.

The two chief points of view from which an estimate of the future of the 2-cycle engine must be made, namely, those of weight per h.p. and fuel economy, merge together finally into a single figure for the weight per h.p. of the power plant, engine and fuel, necessary for a given range of flight. On the question of fuel economy some comparative figures with the 4-cycle Diesel engine are available, and these can be supplemented by certain general conclusions of a fairly definite nature; but, on the other hand, of all the problems connected with aero-engines that of estimating the relative weights of different projected types is the most difficult, and the most unreliable in the result.

It must be remembered that the fully developed aero-engine of to-day has a very large reserve in its margin of safety. Its power/weight ratio is that arrived at after mature experience, during which the design has been conditioned by the need of running for several hundreds of hours without attention. How large this reserve of strength is under the normal conditions of operation is clearly shown by the example of special racing engines in which it has been demonstrated that the power/weight ratio of a normally designed engine can be more than doubled, provided not more than a few hours of working life under these very severe conditions are expected of it. One can at any time take an engine developed to combine a certain power output with a certain life and reliability, and then increase the power substantially at some sacrifice of reliability. This margin of safety required for general use is not calculable. It can only be arrived at by experience, and until time has provided this no man

can say for certain what the weight per h.p. of a new type of engine will be.

The Packard 4-cycle Diesel engine was produced and did a very considerable amount of flying at a weight per rated h.p. of 2·27 lb. Being the first of its type, it might be argued that further development would certainly reduce that figure. On the contrary, that figure was only achieved in the early Packard engines by a combination of very skilful mechanical design and the use of maximum pressures in the cylinders running to more than 1,200 lb. per sq. in. The type was stillborn and shows no sign of future vitality. Other designs of 4-cycle engines have been produced, in this country and elsewhere, in which the mechanical design is more conservative and which could probably be manufactured as power units of the requisite reliability for practical service; but their weights range from 2½ to 3 lb. per h.p. A large amount of research has been carried on in the effort to burn a greater proportion of the air present in the cylinders of high-speed Diesel engines than the present maximum of about 75 per cent., but so far without success. The result is that as compared with B.M.E.P.s in the normally aspirated 4-cycle petrol engine ranging from 120 to 140 lb. per sq. in., those in the Diesel engine range from 90 to 100 at the same piston speeds of about 2,000 ft. per min., and one may hazard the forecast that the normally aspirated 4-cycle Diesel engine suffers, and will suffer, under too heavy a handicap in regard to the M.E.P.s obtainable for its well-proved economy in fuel to be able to redress the balance. Every effort to force up the M.E.P. for the sake of more power per unit of swept volume has ended in a sacrifice of that essential economy which can still put the Diesel engine in a class by itself in regard to fuel consumption.

Generally speaking, one can say that the weight per h.p. of an engine will move up or down with the ratio of the maximum to the mean pressure in the cylinder; but only about 20 per cent. of the total weight of an engine is directly proportional to the maximum pressure, and between the 2-cycle and 4-cycle Diesel engine the difference in the ratio of maximum to mean pressure will not be sufficient to affect the structure-weight appreciably. For a given maximum pressure the 4-cycle engine provides a somewhat higher mean, but the difference is small and may be discounted by the smoother running of the 2-cycle. This is not to be read as an argument that the weight per h.p. of the two types should be equal, but that in a cylinder of a given swept volume the structure-weight would be unaffected by any difference between the gas forces in the 2-cycle and 4-cycle engines. If equal mean pressures could be shown at

equal speeds in the 2-cycle, then the weight per h.p. would, of course, be half that of the 4-cycle.

There are, however, the two points already mentioned which tend to make the weight of a 2-cycle engine greater than that of a 4-cycle of the same swept volume, namely, the long cylinders and pistons which are fundamental to the 2-cycle, although less so in the sleeve-valve type, and the need for having a large scavenge blower capable of giving a pressure of not less than 5 lb. per sq. in. and of supplying 25–30 per cent. more air than remains in the engine cylinders to take part in the combustion. When allowance for these increases in weight have been made, one is then free to compare the probable weights per h.p. of the 2- and 4-cycle engines on the basis of the B.M.E.P.s and piston speeds achieved in either, and the number of working strokes per minute.

As already mentioned, a fairly direct comparison of M.E.P.s in single-cylinder research engines of the sleeve-valve type is possible, for cylinders of the same size, but before making the comparison one must define the conditions in regard to the speed, and the fuel and air supply, and to do so fairly is no easy matter. It is necessary to make a simultaneous comparison of the power output and the fuel economy of the two types, for certain chosen conditions of running, because in any Diesel engine the fuel economy depends largely upon the power developed, and, for any given M.E.P., upon what maximum pressure is allowed in the cylinder. The two types can fairly be compared at the same maximum pressure, and for the rest, the most practical course appears to be to choose two pairs of conditions, the one pair near full power and the other corresponding to a high economy for cruising conditions. Neither full power nor cruising conditions will necessarily be at the same B.M.E.P. nor the same speed in the two types of engine. These will be chosen in each case to suit the type.

The curves in figs. 118 and 119 show the variation of B.M.E.P. with speed, and of fuel consumption with B.M.E.P., observed on two research engines, the one a 2-cycle and the other a 4-cycle, of the same bore and stroke, $5\frac{1}{2}$ in. $\times$ 7 in. Both engines were of Ricardo's sleeve-valve design employing rotational air swirl, and were virtually identical as regards the design of the combustion space.

The scavenging air for the 2-cycle engine was supplied from a separate compressor, and therefore from the gross B.M.E.P. given by the upper dotted curve in fig. 118 a reduction must be made in estimating the net B.M.E.P. available from an engine driving its own compressor. If a compressor of 65 per cent. efficiency is assumed, then the necessary scavenge air at a pressure of 1·5 atm. abs.

would absorb an amount of power equivalent to about 12 lb. per sq. in. in the working cylinder. The lower dotted curve in fig. 118 shows the B.M.E.P. with this allowance made.

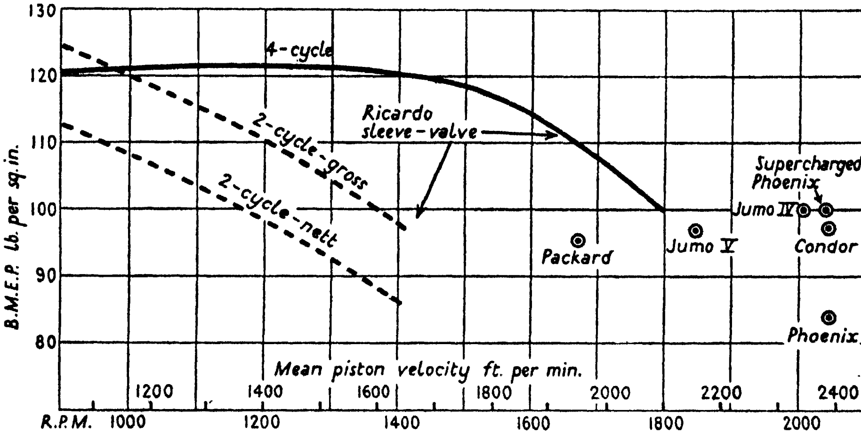

FIG. 118. Full-load B.M.E.P. attained in 2-cycle and 4-cycle Diesel engines of 7 different types.

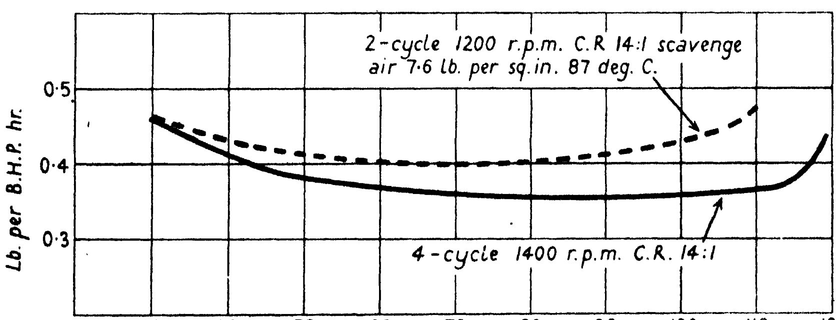

FIG. 119. Comparative fuel consumption per B.H.P. hour of 2-cycle and 4-cycle engines of the sleeve-valve type, and of the same swept volume.

The gross B.M.E.P.s were the absolute maxima attainable, regardless of the rate of fuel consumption, as may be seen for 1,200 r.p.m. by reference to fig. 119. While these highest figures may reasonably be taken in making a comparison of the power output obtainable from 2-cycle and 4-cycle cylinders of the same size, it must be understood that the figures for h.p. per unit of swept volume so obtained are probably higher than could be maintained over long periods of running. With this proviso we may take the net B.M.E.P. for the 2-cycle at full load as 98·5 lb. per sq. in. at 1,200

r.p.m., or alternatively it might be taken as 92·5 at 1,300 r.p.m. If the higher speed is taken the power is only increased by 2 per cent., and this is within the degree of accuracy of any possible estimate on the data available.

To compare with the net B.M.E.P. of 98·5 at 1,200 r.p.m. on the 2-cycle, the 4-cycle can show 100 lb. per sq. in. at 1,800 r.p.m., from which no deductions have to be made. The maximum power available from two engines of the same bore and stroke, therefore, is in the ratio of 1·27 in favour of the 2-cycle. The prospective advantage of the 2-cycle engine is substantial, but very modest when compared with something in the region of 100 per cent. such as might have been looked for on the grounds of there being twice the number of working strokes per revolution of the crankshaft.

It will now be useful to check this comparison by reference to such data as there are upon complete engines. For this purpose it is more satisfactory to introduce the mean piston velocity in ft. per min. in place of the revolutions per min., for then differences of stroke are taken into account. The product (B.M.E.P.) $\times$ (mean piston velocity) when divided by $2 \times 33{,}000$ for the 2-cycle and by $4 \times 33{,}000$ for the 4-cycle engine, gives the h.p. per sq. in. of piston area.

A scale of mean piston velocity for the two research engines has been added at the bottom of fig. 118, and in relation to this scale there have been added to the figure a number of points, representative of three 4-cycle and two 2-cycle engines, all of which have been flown as practical power units, although they may not have been serious competitors of the 4-cycle petrol engine. Comparative figures for these complete engines and for the two research engines have been collected together in table 45, which gives, apart from essential data, the weight per h.p., the h.p. per unit of swept volume, and per sq. in. of piston area, for each engine. The data for the complete engines are based upon figures for the normal rated power in each case.

Several points of interest may be noticed in the figures of table 45. In the first place the Packard engine, in spite of its low h.p. per unit of swept volume, was able to show a weight per h.p. ratio of only 2·27 by reason of its essentially light form (air-cooled radial) and skilful design. The Jumo IV, on the other hand, in spite of a h.p. per unit of swept volume about twice that of the Phoenix, was only 20 per cent. lighter, even when the latter was normally aspirated. In its supercharged form the Phoenix would be almost, if not quite, on a level, when the weight of the starting gear, water, and radiator have been added to the Jumo figure. The comparison reflects the

TABLE 45

Comparative figures of normal rated power output on 4-cycle and 2-cycle compression-ignition engines.

	4-cycle engines				*2-cycle engines*		
	Ricardo sleeve W.C. $5\frac{1}{2}\times 7$	*Phoenix radial 9-cylinder A.C.* $5\frac{3}{4}\times 7\frac{1}{2}$	*Condor in line 12-cylinder W.C.* $5\frac{1}{2}\times 7\frac{1}{2}$	*Packard radial 9-cylinder A.C.* 4·8×6	*Ricardo sleeve W.C.* $5\frac{1}{2}\times 7$	*Jumo IV 2-piston 6-cylinder W.C.* 4·72×8·26	*Jumo V 2-piston 6-cylinder W.C.* 4·14×6·3
Rated h.p.	..	350†	500	225	..	720	500
Engine weight	..	1,066	1,530	510	..	1,775*	1,090*
r.p.m.	1,800	1,900	1,900	1,950	1,370	1,700	2,050
Piston velocity	2,100	2,380	..	1,950	1,600	2,340	2,160
B.M.E.P.	100	83·5	97	95	88	96·5	97
h.p./sq. in. of piston area	1·59	1·50	1·75	1·41	2·13	3·43	3·10
h.p./100 cu. in. of swept vol.	22·7	20·0	23·4	23·6	29	41·5	49
Weight per h.p.	..	3·05†	3·06	2·27	..	2·47*	2·18*

* These are figures for dry weight, without starting gear.

† Power when normally aspirated. The same Phoenix engine, when fitted with a supercharger giving an induction pressure $3\frac{3}{4}$ lb. per sq. in. above atmospheric, gave 420 h.p. and a weight per h.p. 2·61 lb. at normal r.p.m.

100 cu. in. = 1·64 litres.

essentially heavy character of the Jumo design, with its two six-throw crankshafts and crankcases, only partially counterbalanced by the absence of any cylinder-heads and valve-gear.

The very high output of the Jumo engines per unit of swept volume—1·75 times the best of the 4-cycle engines, as compared with the estimated ratio of 1·27 between the Ricardo engines—appears to indicate that with their long cylinders they are able to secure an efficient scavenge and high M.E.P.s with a considerably lower scavenge pressure than is possible in Ricardo's sleeve-valve cylinders with a single piston; and in consequence there is less difference between the gross and net M.E.P. Exact data are not available, but it appears that the Jumo engines use a scavenge pressure of about 3–4 lb. per sq. in. only, and, with this, can obtain a gross B.M.E.P. of the order of 110 lb. per sq. in. at piston speeds substantially higher than in the sleeve-valve cylinder.

The high scavenging efficiency of the 2-piston cylinder here indicated, and previously foreshadowed by Bird's experiments with water, has been confirmed by Neumann[74] with a single-cylinder engine of the Junkers type. In these experiments a scavenging efficiency of more than 95 per cent. was achieved in a cylinder of

the same length/diameter ratio as in the Jumo IV engine. Besides his experimental results, Neumann's paper contains a full analysis of the pressure and temperature conditions during the scavenging process.

As regards fuel economy, fig. 119 shows that the 2-cycle sleeve-valve cylinder gives a minimum figure of 0·4 lb. per B.H.P. hour at a B.M.E.P. corresponding to cruising conditions, as compared with 0·36 in the 4-cycle. Moreover, here again allowance must be made for the work to be spent on the scavenge blower. This should not be equivalent to more than about 7 lb. per sq. in., or say 8 per cent. of the gross power under cruising conditions, and the net cruising consumption should therefore be 0·435 lb. per B.H.P. hour.

The fuel consumptions of the complete engines agree well with the research cylinder figures, except that here again the Jumo engine appears to outstanding advantage when compared with the sleeve-valve 2-cycle. As against the estimated cruising consumption for the latter of 0·435 lb. per B.H.P. hour, the Jumo's consumption ranges rather with the 4-cycle engines. Under cruising conditions both the Phoenix and the Condor engines have shown consumption figures of 0·37 to 0·38 lb. per B.H.P. hour, and the Jumo IV a figure as low as 0·36. The Jumo V is reported to have completed a 50-hour run at only just under 500 B.H.P. with a fuel consumption of 0·38. This, combined with a weight per B.H.P. (at maximum power) of 1·9 lb., would bring the engine well into the field as a competitor with the modern petrol engine.

The fundamental advantage held by the 2-piston engine, of having no cylinder-heads to absorb waste heat, has already been mentioned. In art. 61 (i) it was estimated that 5 per cent. of the total heat supply in a 4-cycle engine was lost to the cylinder head before the expansion had seriously begun; and that in a cylinder of expansion ratio 5 : 1 this would mean a loss of 2 per cent. in the figure for the thermal efficiency. In a Diesel engine, with a sleeve-valve cylinder and air-swirl, the rate of heat loss must be very high during combustion, and the fraction of the total heat supply lost to the head would be above rather than below 5 per cent. The higher expansion ratio in the Diesel cycle, moreover, means that any heat saved could be converted into useful work at an efficiency of about 45 per cent. as compared with 40 per cent. in the petrol engine.* The figure of 2 per cent. arrived at for the petrol engine as the probable increase of thermal efficiency due to a complete suppression of the head losses might very well be increased to 3 or 4 per cent., therefore, in estimating the fundamental advantage of the Jumo type of engine

* Refers to expansion process only, see art. 61 (i), p. 229.

over the single-piston cylinder in respect to thermal efficiency. Now an increase of 4 per cent. on a brake thermal efficiency of 35 per cent. would mean a reduction of the fuel consumption per h.p. from 0·435 to 0·39, and would account for a large part of the discrepancy between the observed fuel economy of the 2-piston and 1-piston types of 2-cycle cylinders.

Whether, in the light of this conclusion, the sleeve-valve form of 2-cycle engine can ever do so well as the 2-piston type, is an interesting question. There seems to be little doubt that, besides the essential disadvantage in thermal efficiency, more of the gross h.p. will have to be absorbed for scavenging purposes; but, on the other hand, the sleeve-valve design would be perfectly adapted for development in the radial form, which is, of course, impossible for the 2-piston type of cylinder. The advantage to be gained from the essential lightness of the radial form of engine may possibly counterbalance the benefit of the good scavenge in the 2-piston type so far as weight per h.p. is concerned; but it is difficult to see how the single-piston cylinder can ever equal the 2-piston type on the score of fuel economy.

XII

ANALYSES OF COMPLETE ENGINE PERFORMANCE

No book on the aero-engine would be complete without some account of the data now available on the performances, in regard to power output and fuel economy, achieved by representative engines on the test bench and in the air. In any such account, however, the writer is faced by a serious difficulty. To be readily apprehended, the available data should be expressible in formulae for general application, but this is possible only to a very limited extent. For the most part the information must be set down in the form of numerical tables of results, tedious to the reader, and carrying on the face of them no general significance whatever.

Where possible in the following three chapters the experimental results have been reduced to the form of generally applicable equations, as in the articles on mechanical friction and pumping loss, and on the variation of engine power with height. For the rest, if the reader will have patience with the inevitable arithmetic he should find it possible to extract from it a representative picture of modern aero-engine performances.

ART. 69. *Power variation with external air temperature.*

In art. 12 the method of estimating the performance of an aeroplane in the standard atmosphere was given, at first on the assumption that the power of an engine is directly proportional to the density of the air in which it works; and thereafter certain corrections were given to allow for a power variation more closely in accordance with actual experimental results. It was stated that no serious error would be involved by taking the B.H.P. of the engine as proportional to the product of the air density and the square root of its absolute temperature. If the engine power is taken simply as proportional to the density ratio σ, the error will be an over-estimate of the power by 4 per cent. at 10,000 ft.

In the present chapter the factors which influence the variation of power of an engine will be examined in detail, and in doing so it is convenient to deal separately with the influence of the external air pressure and temperature upon the 'indicated' power of the engine, and upon the mechanical friction and pumping losses.

It was shown in art. 19 that when a cylinder was supercharged the I.H.P. was very closely proportional to the weight of fresh charge per cycle. This must be so, indeed, unless there is some factor which

upsets the combustion and prevents the same amount of heat from being generated per lb. of air drawn in per cycle. In the supercharging experiments the weight of fresh charge per cycle was not proportional to the density of the air supply, because, the residual gas being always at atmospheric pressure, there was a progressive increase of the volumetric efficiency of the cylinder as the supercharge pressure was increased.

When a non-supercharged engine rises from ground-level the pressures on the inlet and exhaust sides diminish together, and if there were no changes of temperature of the ingoing air during its passage into the cylinder the charge weight, heat generated, and I.H.P., or I.M.E.P., would be closely proportional to the external air density. The fact that they are not so is due to the changes of temperature of the combustible mixture as it enters, caused by evaporation of the fuel and by the heat picked up from the induction system and cylinder walls before the inlet valve closes.

The air from the moment when it enters the carburettor becomes associated with liquid petrol in process of evaporation, and the necessary heat for vaporizing the petrol is drawn from the air-stream and from the metal parts of the carburettor. The carburettor body and the air-stream are cooled down rapidly, the latent heat of petrol being sufficient to lower the temperature of a 1 : 14 mixture of fuel and air by 20° C. or more. From the carburettor onwards the fuel-air mixture is warmed by contact with the walls of the induction system, and this warming becomes more rapid as each element of the mixture approaches the inlet valve and passes into the cylinder. It is the average temperature of the air in the cylinder just as the inlet valve closes that settles the weight of air drawn in per stroke.* This average temperature cannot be measured directly, but the indirect evidence supports the conclusion that when the ignition timing is properly adjusted the I.H.P. is always very nearly proportional to the weight of fresh charge taken in per cycle, just as it was in the single-cylinder supercharging experiments of Chapter III.

By the time the indrawn mixture has come into contact with the cylinder walls and piston, and as a rule some time before the inlet valve has closed, sufficient heat will have been picked up to complete

* There would normally be a small drop of pressure of about 0·5 lb. per sq. in. at the carburettor (at full throttle), and a further drop of 3–4 lb. per sq. in. at the inlet valve during most of the suction stroke. The inlet valve does not close, however, in a high-speed engine until some 40–50 deg. after the end of the stroke, and experiment has shown that this gives time for the pressure in the cylinder to rise to that in the manifold just as the valve closes. This question will be dealt with more fully when considering the pumping losses in the next article.

the evaporation of the fuel, and any further heating will all go to raise the final mixture temperature, and so to lower the volumetric efficiency and the power output. A change of external air temperature will very much alter the progress of the fuel evaporation, as illustrated below in fig. 120, and it will also affect the total amount of heat received by the mixture during its passage through the induction system, since the metal temperatures will be substantially the same

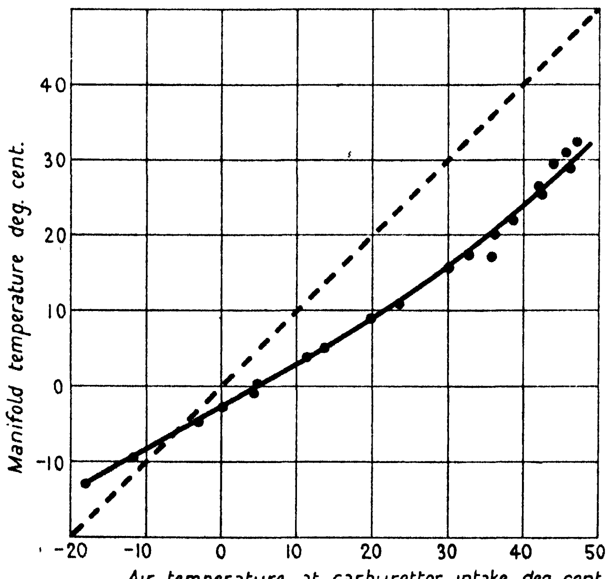

FIG. 120. Variation of mixture temperature in the manifold with external air temperature.

under all conditions. The effect of the fuel, moreover, will be to accentuate the amount of heat picked up by the ingoing mixture when the external air temperature is low. This arises from the fact that more of the fuel is evaporated from the metal surfaces under those conditions, and the rate of heat absorption by the mixture as a whole will then be greater than if the hot metal were swept by dry air only.

The experiments illustrated in fig. 120, which were made upon a 12-cylinder Liberty aero-engine,[57] show that the mixture temperature as measured in the manifold close to an inlet valve was the same as the external air when this was $-7°$ C. The heat from the carburettor body and the manifold up to the point of measurement was then just sufficient to balance the small amount of evaporation which had taken place. During a rise of the external air temperature from $-18°$ to $+49°$ C. the air in the manifold rose only by $44°$ C. from

-13° to $+31^\circ$, because at the higher temperatures more fuel was being evaporated in the carburettor and at the same time less heat was being picked up from the metal surfaces. In the mixture at -13° the petrol must have been practically all in the liquid state and the rate of heat absorption by the mixture during the later stages will have been the more rapid.

By working back from the measured volumetric efficiency of the Ricardo E. 35 engine, with heat supplied at the intake to an amount to imitate the manifold conditions on a multi-cylinder engine, it has been estimated that with an external air temperature of 15° C. there must be a total rise of temperature of the ingoing charge of about 105° when the compression ratio is 5 : 1.

It is of some interest, although the estimate can only be a rough one, to calculate from this the mixture temperature at the closing of the inlet valve for the two extreme conditions shown in fig. 120. A reasonable assumption would be that the further rise of temperature above that recorded in the manifold would be proportional to the difference between that temperature, as given in fig. 120, and some mean temperature for the piston and cylinder walls, say 200° C.

According to fig. 120, when the external air temperature was $+15^\circ$ C. that in the manifold would be $+5^\circ$ C. As compared with an average difference of 195° under these conditions between the metal and the in-going mixture, the differences under the two extreme conditions of fig. 120 would be 213° and 169°. The rises of temperature after leaving the manifold, therefore, may be roughly estimated as 125° and 100°, as compared with 115° in the E. 35 engine. The final temperatures would then be 112° and 131°, except that evaporation of nearly all the petrol has to be allowed for in the low temperature condition and none in the other.* Allowing for this we arrive at 92° and 131° as the final mixture temperatures before compression in the two conditions when the external air temperatures were -18° and $+49^\circ$. The point of interest is that the ratio of the two mixture temperatures abs., 0·9, is very nearly the square root of the ratio of the air temperatures, 0·79; for in the next article it will be shown that experiments on a variety of engines and over the range of air temperatures from -20° to $+40^\circ$ C. have shown that in every case the I.H.P. varied very nearly in proportion to $1/\sqrt{T}$. The greatest divergence was less than 2 per cent. and for the most part it was 1 per cent. or less.

The observed variation of power with inlet air temperature can therefore be explained in terms of cooling by petrol evaporation and

* The uniform cooling of 15° shown in fig. 120 for all air temperatures above 30° C. is evidence that above this point there was complete evaporation in the manifold.

heating of the mixture during entry to the cylinder; and it is consistent with a variation of I.H.P. in proportion to the mixture temperature abs. at the moment the inlet valve closes.

ART. 70. *Engine trials with varied external air temperature.*

In all trials to test the effect of a change of external air temperature it is essential to maintain a known and nearly constant fuel-air ratio; otherwise the external and internal air temperatures need have no

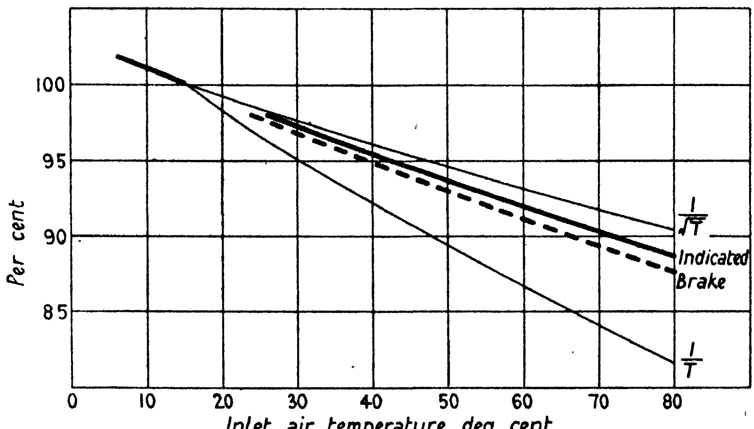

FIG. 121. Variation of I.H.P. and B.H.P. of a 12-cylinder water-cooled engine with inlet air temperature, shown as a percentage of the value at 15° C. Bore and stroke 5 in.×7 in. Compression ratio 5·4. Range of speeds 1,400–1,800 r.p.m.

consistent relation to one another, because of different degrees of cooling by evaporation. In the trials of which the results are given below the tests were made over a series of fuel-air ratios under each set of conditions, both air and fuel consumptions being measured, and the I.H.P. was then chosen as that obtained with the weakest mixture consistent with maximum power. The I.H.P. was in each case obtained by adding to the observed B.H.P. the power needed to motor the engine under the same conditions as in the power test.

In a set of trials[58] upon an 8-cylinder water-cooled aero-engine, in which the compression ratio was varied from 5·3 to 8·3 and the inlet air temperature from −20° to +40° C., it was found that throughout all the conditions the I.H.P. at full throttle varied almost exactly in proportion to $1/\sqrt{T}$, where T is the external air temperature abs., other conditions being constant. The variations of the other conditions included ranges of speed from 1,400 to 1,800 r.p.m.; throttle positions corresponding to full, ¾, and ½ load; and external air pressures corresponding to heights of 5,000, 15,000, and 25,000

ft., besides ground-level. The air measurements showed that over the whole range of conditions the I.H.P. at any one compression ratio was closely proportional to the weight of mixture drawn in, and that this was proportional to $1/\sqrt{T}$ when the external pressure was constant and the temperature not above 40° C.

In a further set of trials,[59] which included both a water-cooled 12-cylinder Liberty and an air-cooled radial engine, the air temperatures were carried up to +80° C., and it was found that over this

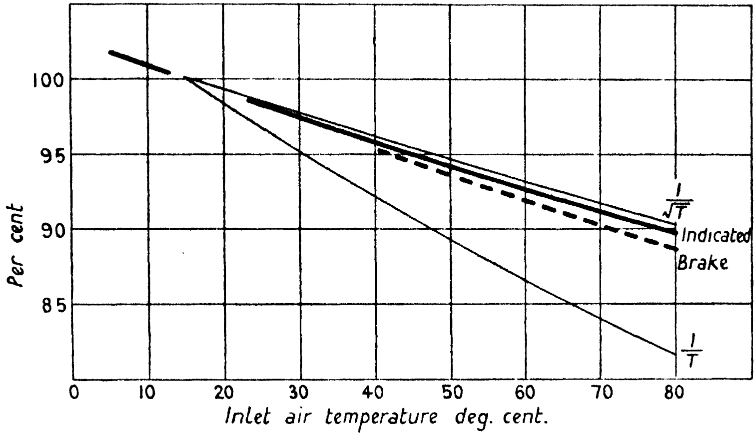

FIG. 122. Variation of I.H.P. and B.H.P. of a 9-cylinder air-cooled radial engine with inlet air temperature, shown as a percentage of the value at 15° C. Bore and stroke 4·72 in.×5·12 in. Compression ratio 5·3. Range of speeds 1,400–2,000 r.p.m.

higher range of temperatures the decrease of power was rather more rapid than in proportion to $1/\sqrt{T}$. The power variation for these two engines, both indicated and brake, is shown in figs. 121 and 122 as compared with variations proportional to $1/\sqrt{T}$ and $1/T$. In both engines the variation is much nearer the former than the latter, the greatest deviation from the $1/\sqrt{T}$ rule for the I.H.P. being about 2 per cent. for the water-cooled engine at an air temperature of 80° C.

The figures for the friction, or lost, horse-power (L.H.P.) obtained in these trials for deriving I.H.P. from the B.H.P. showed for the most part a small diminution with a rise in the external air temperature. The factors which influence the L.H.P. are complex, and since an understanding of them is important they will now be examined in detail in the next three articles.

ART. 71. *The lost horse-power. Full-throttle conditions.*

The ultimate object of all analyses of engine performance is to enable the changes of torque, Q_E, and the fuel consumption per B.H.P. hour to be predicted for changes of throttle position, speed,

and atmospheric conditions; and to understand the underlying causes of the variations. There are, moreover, two sets of conditions under which the study of engine performance is of particular interest, namely, the variation of torque at full throttle with change of height in the standard atmosphere, and secondly, the variation of fuel consumption with throttle position and speed in level flight.

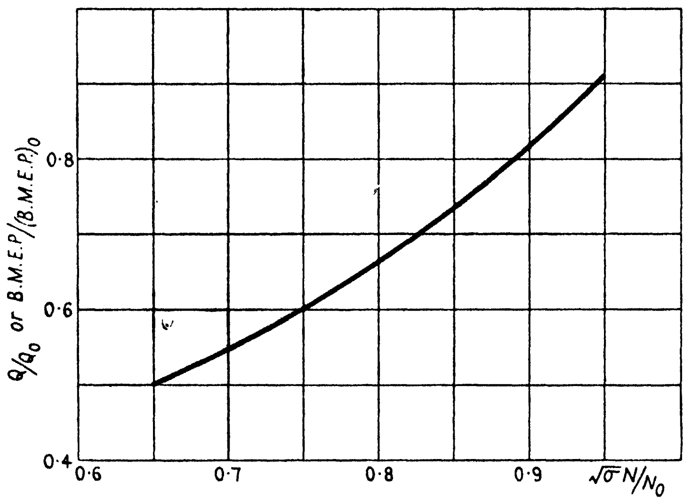

FIG. 123. Relation between B.M.E.P. and r.p.m. in level flight which holds for all heights and for all aeroplanes of normal parasite drag. N_0 is the maximum permissible r.p.m. of the engine and Q_0 is the full-throttle torque, at ground-level for a normally aspirated engine and at the 'rated altitude' for a supercharged engine.

N.B. The airscrew would be designed so that Q_0 produced N_0 in level flight.

From each of these points of view the variation of L.H.P. is of much importance; and since it will be altered by a change of any one of the variables speed, throttle, cylinder wall temperature, and air pressure and temperature, a complete study of the changes of L.H.P. or L.M.E.P. under all conditions would be extremely complex. Fortunately the variables are not all independent, and this enables the problem to be restricted and simplified. It has been shown,[60] for example, that in level flight there is a unique relation which holds between engine speed and engine torque (or B.M.E.P.) for any height and, with sufficient accuracy, for practically all types of aeroplane. When considering fuel consumption in flight, therefore, we are only concerned with the variation of the L.H.P. at various heights over a certain range of related speeds and throttle settings. And again, when considering the variation of L.H.P. with height we are only concerned with the values at full throttle and for air pressures and temperatures which are related as in the standard atmosphere.

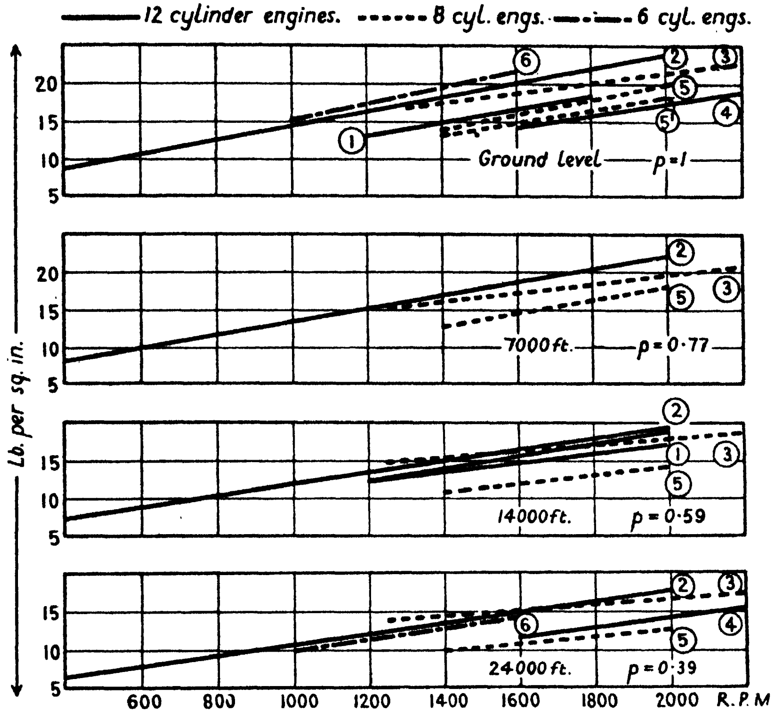

FIG. 124. Variation of lost power with speed and atmospheric pressure, expressed as M.E.P. on the pistons. Compression ratios between 5·3 and 6·5. Jacket outlet temperature 70° C.

In what follows, the experimental results will first be presented so as to bring out the independent effects of the different variables, and later on the required conclusions will be drawn for the restricted conditions applicable to engines operating in the air. Before proceeding with the experimental details it will be appropriate to give here the relationship between speed and B.M.E.P. in level flight referred to above. This is shown in fig. 123, the B.M.E.P. as a fraction of its full-throttle value and the speed as a fraction of the maximum permissible speed of the engine. The curve given applies to a geared engine in an aeroplane of normal parasite drag. For variations from this mean curve the original reference should be consulted.[60]

The L.M.E.P. in a throttled engine, applicable over the range of speeds in level flight, will be discussed later. Taking at first only observations at full throttle, there are given in fig. 124 the results of L.H.P. measurements, expressed as L.M.E.P., on 6 different water-cooled engines,[61] 3 of 12 cylinders, 2 of 8, and 1 of 6, over ranges of speeds from about 1,000 to 2,200 r.p.m. and under pressure

conditions corresponding to heights in the standard atmosphere of 7,000, 14,000, and 24,000 ft. And in fig. 125 are the results of similar trials[47] with a modern 12-cylinder water-cooled engine carrying a geared supercharger capable of maintaining ground-level pressure up to 9,600 ft. The identification figures 5 and 5′ in fig. 124, as also 6 and 6′ in fig. 126, refer to the same engine with different compression ratios.

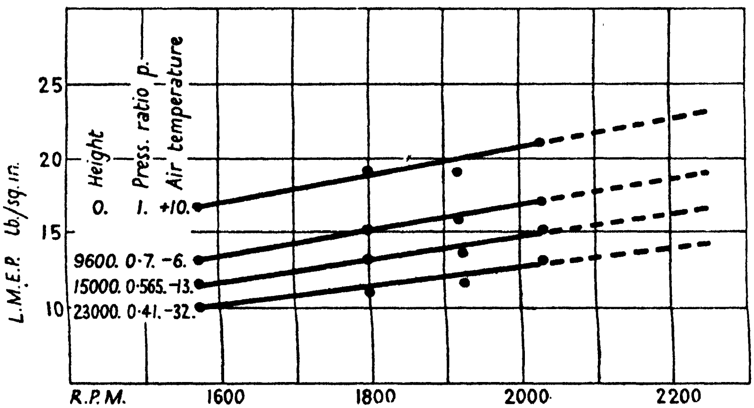

FIG. 125. Variation of L.M.E.P. with speed in a 12-cylinder water-cooled engine having a gear-driven supercharger maintaining ground pressure to 9,600 ft. at 2,250 r.p.m. Full-throttle observations except at ground-level. Normal engine speed 2,250. Bore and stroke 5 in. × 5½ in. Compression ratio 6 : 1.

It is not proposed to analyse these results at present, but it may be remarked in passing that, considering the wide variety of engines covered by the tests, they are satisfactorily consistent. They all show values of the L.M.E.P. with a nearly linear increase with speed that is closely the same for all the engines, 1 lb. per sq. in. per 100 r.p.m. The values diminish with height at the normal full speed of the engines at the rate of 5 lb. per sq. in. in 24,000 ft. This is true equally in the supercharged engine of fig. 125 for heights above 9,600 ft. The larger difference between the motoring loss there and at ground-level is because at ground-level the engine was partially throttled so as to give the same pressure in the induction pipe as at the rated altitude of 9,600 ft. Apart from the difference of the inlet air temperature at the two heights, therefore, and its slight effect on the supercharger, the increase in the motoring loss at ground-level shown in fig. 125, over that at the rated altitude, corresponds simply to the difference of the pressure on the exhaust and inlet sides of the engine, about 4½ lb. per sq. in.

In fig. 126 the variation of L.M.E.P. with external air pressure

is shown for 11 different normally aspirated water-cooled engines of 6, 8, and 12 cylinders, and of compression ratios between 5·3 and 6·3, all but one at the single speed of 1,600 r.p.m. The engines numbered 1 to 6 are the same as those in fig. 124. The inlet air temperature in all but two of the tests was nearly constant at 15° C., and the jacket-water outlet at 70° C. Under these conditions the

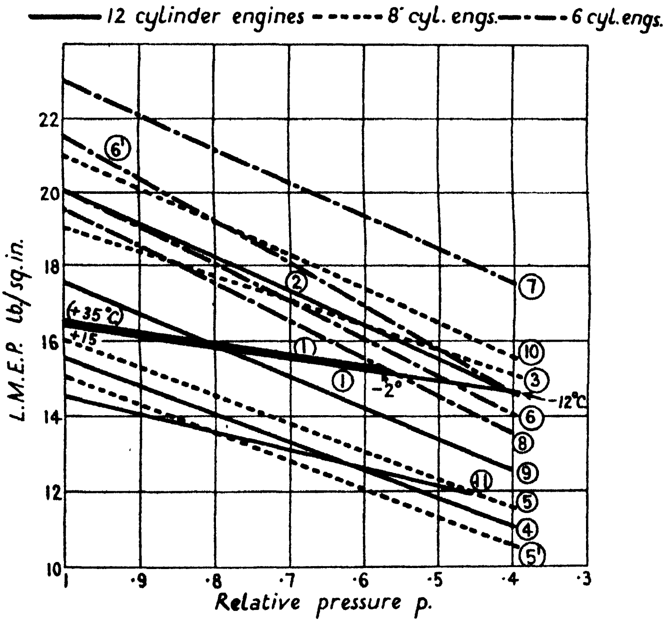

Fig. 126. Variation of L.M.E.P. with atmospheric pressure at constant speed 1,600 r.p.m. and nearly constant inlet air temperature about 15° C. Compression ratios between 5·3 and 6·5. Jacket water outlet temperature 70° C.

measured values of the L.M.E.P. at any given external pressure vary over a range of 7 or 8 lb. per sq. in., but this reflects differences of piston design, of valve area, and especially differences in the number of cylinders. The highest lines in fig. 126 all refer to engines of 6 or 8 cylinders, for in these the power lost in the main bearings and auxiliaries, like water and oil pumps and ignition gear, has to be divided between a smaller number of cylinders than in the 12-cylinder class.*

* During tests of this character carried out at ground-level the engine itself should, for strict accuracy, be in a low-pressure chamber, besides having the pressure in the inlet and exhaust manifolds maintained at the proper value. When this is not done there will be a slight leakage of air past the engine valve stems, but comparative tests under the partial and the truly representative altitude conditions have shown that the difference of engine performance was too small to be measured.[67]

It is of interest that the test of engine 11, the Curtiss 'D. 12' of 12 cylinders, was at 2,000 r.p.m. and yet the engine showed an L.M.E.P. as low as any. The 'D. 12' was designed about ten years later than most of the rest in fig. 126, and the lower motoring loss is probably due to the improvement of the piston, and particularly of the valve port, design in the interval. Enlarged and improved valve ports lead to a lower pumping loss, especially at low heights. It will be noticed that engine 11 showed a less rapid fall of L.M.E.P. with atmospheric pressure at constant speed than the other tests with a constant air temperature; and when p was less than 0·62 the total L.M.E.P. was greater than that of engine 4 at 1,600 r.p.m.

The pair of straight lines in fig. 126, between the ends of which the inlet air temperature varied from +35° to −2° C., and from +15° to −12° C., show a markedly smaller variation of L.M.E.P. with p than the rest. The data from these tests is of special interest and is given in greater detail in tables 46 and 47. The results will be

TABLE 46

Variation of losses with speed and barometric pressure, as measured by a motoring test of a 'Liberty' 12-cylinder water-cooled aero-engine.

	Ground-level		*15,000 feet*	
	Jacket temperature 75° C. *Oil: inlet 34·5°; outlet 69·5°* *Relative pressure p = 1* *Air temperature 35° C.*		*Jacket temperature 73° C.* *Oil: inlet 34°; outlet 69°* *Relative pressure p = 0·57* *Air temperature 0° C.*	
Speed r.p.m.	*L.H.P.*	*L.M.E.P.*	*L.H.P.*	*L.M.E.P.*
1,200	33	13·2	30	12·0
1,400	43	14·8	40	13·8
1,600	55	16·5	51	15·3
1,800	69	18·5	61	16·3
2,000	(90)	21·6	71	17·1

TABLE 47

Variation of losses at constant speed of 1,600 r.p.m. and varying barometric pressure, as measured by a motoring test. Same engine as table 46.

Barom. press. relative to ground-level p	*Air temperature*	*Relative density* σ	*L.H.P.*	*L.M.E.P.*
1	15°C.	1	55	16·5
0·84	5°	0·87	53·5	16·1
0·70	−3·5°	0·75	52	15·6
0·575	−10·5°	0·63	51	15·3
0·475	−13·5°	0·525	50	15·0
0·39	−11·5°	0·43	49	14·7

referred to again in the next article, in which it will be shown that about half of the total L.M.E.P. may be expected to vary in the ratio $1/\sqrt{T}$, where T is the external air temperature. In most of the other observations shown in fig. 126 the average rate of fall of L.M.E.P. is substantially the same, and may be expressed in the form $8(1-p)$ lb. per sq. in., where p is the ratio of the pressure to the normal ground-level value.

Some observations upon the rate of variation of the loss with inlet

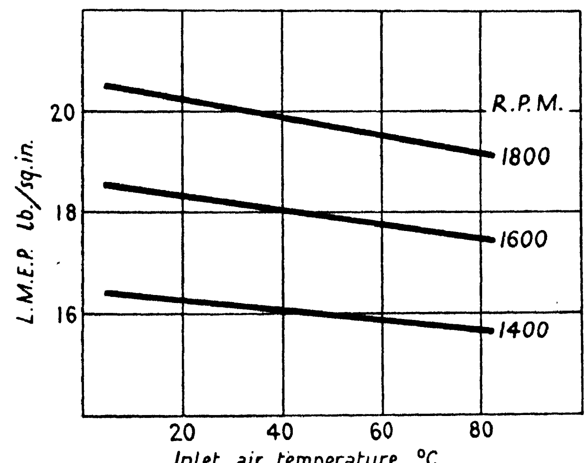

FIG. 127. Variation of L.M.E.P. with inlet air temperature. 12-cylinder water-cooled Liberty engine. Compression ratio 5·4. Bore and stroke 5 in. × 7 in. Normal full speed 1,800 r.p.m.

air temperature, under constant pressure conditions, are shown in fig. 127 for a 12-cylinder engine.[59] Unfortunately the results of only one set of tests are available.

The importance of keeping exact control of the cylinder-wall temperature when making observations of L.H.P. is brought out in fig. 128, which shows the observed variation of L.M.E.P. with jacket-water outlet temperature for four engines at different speeds. The rate of decrease is about 1 lb. per sq. in. for a rise of jacket temperature of 10° C., but at the highest speeds the effect may be more marked. With truly constant cylinder-wall conditions the whole change of L.M.E.P. with air pressure or temperature must be due to changes of the pumping loss, but a change of oil viscosity in the cylinder may easily mask the true changes of pumping loss which the experiment was designed to measure.

A point of some interest in connexion with jacket-water temperature has been brought out in a recent report[67] in which measurements

are given of the B.H.P. and the motoring loss (L.H.P.) at ground-level and at the equivalent of 10,000 and 20,000 ft. on a Curtiss D. 12 engine (no. 11 of figs. 126 and 128). The measurements were made over a wide range of jacket-water outlet temperatures, from 30° to 100° C., and the linear fall of L.M.E.P. was shown to hold over the

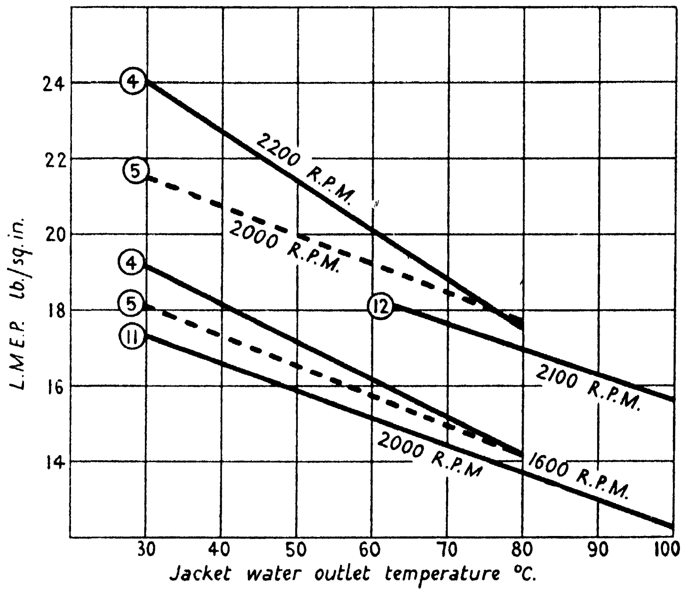

FIG. 128. Variation of L.M.E.P. with jacket-water outlet temperature. Engine (4): 12-cylinder. Bore and stroke 4·5 in. × 6 in. Compression ratio 5·3. Engine (5): 8-cylinder. Bore and stroke 5·5 in. × 5·9 in. Compression ratio 5·4. Engines (11) and (12): 12-cylinder. Compression ratios 5·6 and 5·8 (see also table 38).

entire range at all heights. Although the mean value of the L.M.E.P. was about 2½ lb. per sq. in. less at 20,000 ft. than at ground-level, the reduction of it by a rise of the jacket temperature from 70° to 100° C., also 2½ lb. per sq. in., was the same at the two heights, because, of course, the change was due to the effect of the cylinder-wall temperature upon the mechanical friction, which was virtually unaffected by altitude.

The point of special interest brought out by these tests was that above the equivalent of about 10,000 ft. a rise in the jacket-water temperature can produce an increase of B.M.E.P. instead of the usual fall, following the fall of volumetric efficiency and I.M.E.P.; the increase being due to the undiminished effect of the rise of temperature on the mechanical friction, while its numerical reduction of the

I.M.E.P. is smaller, in proportion to the lower gas pressures at high altitudes.

The fall of I.M.E.P. for the above change of jacket temperature was only 2 lb., from 67½ to 65½, at 20,000 ft. as against 6 lb., from 153 to 147, at ground-level. Owing to the reduction in the L.M.E.P. of 2½ lb. by the hotter jackets, the B.M.E.P. at 20,000 ft. rose by about 1 per cent. for the 30° difference. Unfortunately at high altitudes, where B.H.P. is to be gained by an increase of cylinder temperature, one is limited by the boiling-point of water to a jacket temperature of about 80° C. at 15,000 ft., and 70° C. at 25,000 ft.; and there is here an argument in favour of a high boiling-point liquid, such as ethylene glycol, for aeroplanes with liquid-cooled engines designed to fly at great heights. The level at which the change occurs from a loss to a gain of B.H.P. by an increase of the cylinder jacket temperature is 9,000–10,000 ft.

ART. 72. *The lost horse-power in a throttled engine.*

Before summarizing the available data upon the variation of the L.M.E.P. in a throttled engine it will be well to indicate in a general way the influence of the induction manifold pressure and the valve timing upon the pumping loss, and why the relationship differs in a single- from that in a multi-cylinder engine.

In a single-cylinder engine, with comparatively little capacity between the throttle and the inlet valve, there will, at small throttle openings, be a large variation of pressure in the induction manifold during each suction stroke, from near atmospheric down to the amount of the cylinder pressure when the inlet valve closes. In a multi-cylinder engine, on the other hand, where a number of cylinders draw in turn from a single manifold, there will be much less pressure variation during the suction stroke. The cylinder pressures in this case will drop to slightly below the nearly steady manifold pressure early in the stroke, whereas in the single-cylinder engine there would be a steady drop from near atmospheric, both in the manifold and in the cylinder, throughout the stroke.

A further difference worth pointing out is that when a multi-cylinder engine is motored round with the throttle partly closed the pressure in the cylinders, when the inlet valves first open, will be well above that in the manifold, and the result is that considerable blow-back occurs. In the single-cylinder the manifold pressure will have had time to rise during the exhaust stroke so as more nearly to balance the cylinder pressure when the inlet valve opens. By reason of the more rapid fall of pressure in the cylinder at the beginning of

the suction stroke, a multi-cylinder engine may be expected to show a larger motoring loss per cylinder, for the same speed and valve timing, than a single cylinder of the same dimensions.

The curve *ABCD* in fig. 129 has been sketched in to show the approximate pressure variation in a multi-cylinder engine during suction and compression when it is throttled so that the average manifold pressure is 9·5 lb. per sq. in., the outside pressure being normal atmospheric. Although the negative work during suction and compression is limited to the area (1), further negative work of

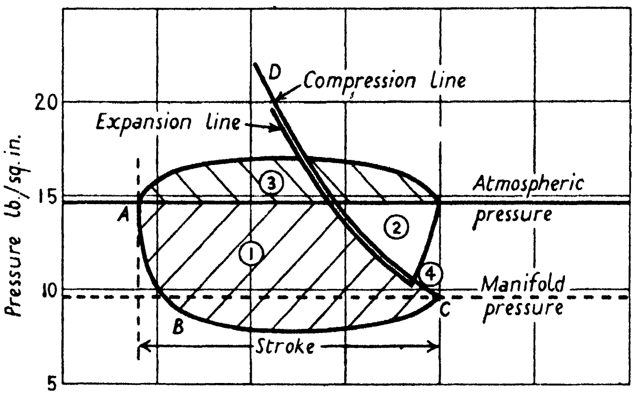

FIG. 129. Showing the indicated pumping loss in a throttled multi-cylinder engine when motored round.

area (2) is performed during the expansion stroke, so that the total indicated pumping loss during a cycle is the sum of the two areas (1) and (2) together with the area (3) representing the exhaust stroke. The meaning of the area (4) and of the line dividing it from (2) is that when the exhaust valve opens, as it normally does, well before the end of the expansion stroke (50–60 deg. early in a high-speed engine), the pressure in the cylinder will rise to that outside the exhaust valve; and the negative work will be reduced in consequence by the area (4). The total negative work, representing the pumping loss, will clearly increase as the manifold pressure is lowered by throttling, whether the engine be normally aspirated or supercharged, and the results of such tests as are available indicate that the increase in the total power taken to motor the engine at any given speed is a linear function of the decrease in manifold pressure.

In fig. 130 are shown the observed values[62] of the total L.M.E.P. at two speeds during the motoring tests of an 8-cylinder water-cooled engine, no. 5 of figs. 124 and 126, as the manifold pressure was reduced from atmospheric down to 2 lb. per sq. in. by throttling.

The rate of increase varies slightly with the speed, and may be expected to vary somewhat, also, from engine to engine according to the valve timing and valve areas; but great accuracy is not required, and it is probable that at the same fraction of the normal full speed the observed increase, from 15·5 to 20 lb. per sq. in. at 1,600 r.p.m., when the manifold pressure was dropped from atmospheric to 6 lb. per sq. in., can be taken as a basis for calculation for any normally

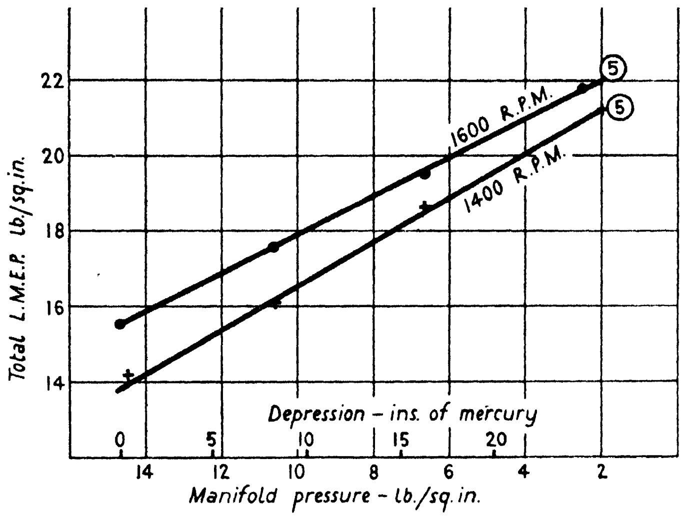

FIG. 130. Variation of total motoring loss in an 8-cylinder engine at different throttle openings. Normal full speed 2,000 r.p.m. Air inlet temperature 15° C. Engine (5) of figs. 124 and 126.

aspirated engine. The whole of the increase at constant speed must be put down to an increase in the pumping loss, for there is no reason why closing the throttle during a motoring test should affect the mechanical friction to an appreciable degree.

The same characteristics are evident in the results[47] shown in fig. 131 for a 12-cylinder engine with a gear-driven supercharger capable of maintaining ground-pressure up to 9,600 ft. These tests cover three heights besides ground-level, and a series of throttle openings at each of three speeds at each height. Corresponding to the observations of total L.M.E.P. at each height there is a vertical line of the same type (full, dotted, &c.) which gives the pressure in the exhaust system at that height.

The interpretation of these results is complicated by the fact that the values of the L.M.E.P. represent the total power to motor the engine, which included the power absorbed in driving the super-

charger, and this would be altered by each one of the variables speed, height, and throttle setting. For the present all that need be pointed out is that the exceptionally high values of the L.M.E.P. under ground-level conditions were due to the engine having been throttled down even when the manifold pressure was atmospheric. The pressure ratio given by the supercharger at an engine speed of 2,030

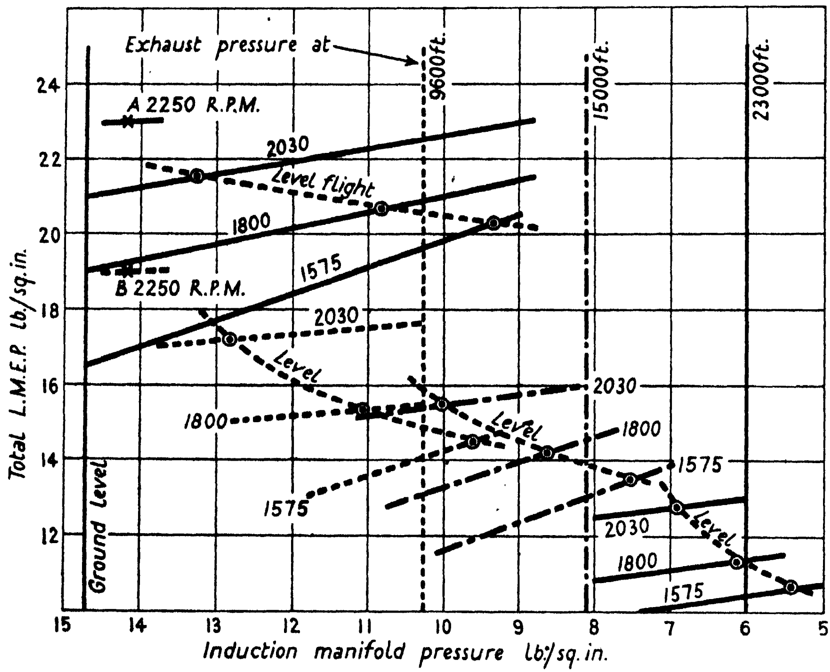

FIG. 131. Variation of total L.M.E.P. in a 12-cylinder engine of rated altitude 9,600 ft. at 4 speeds and various ranges of throttle openings.

Points *A* and *B* estimated from fig. 125.

r.p.m. would be about 1·20, so that the observed L.M.E.P. of 21 lb. per sq. in. included the power to deliver the full air supply, about 45 lb. per min., after compressing it from 12 to 14·7 lb. per sq. in.

There was insufficient power available to motor the engine at its full normal speed of 2,250 r.p.m., but the estimated values of the L.M.E.P. at full throttle (from fig. 125) are shown at *A* and *B* in fig. 131, for ground-level and the rated height respectively. Corresponding to each of the three lower speeds shown in fig. 131 there would be a definite B.M.E.P. for level flight at each altitude, in accordance with the curve of fig. 123, and to give these B.M.E.P.s certain definite manifold pressures would be necessary. As a matter

of interest, and to indicate the use that can be made of the data in fig. 131, there are marked on each set of lines the points corresponding to level flight at each altitude. The important point to notice is that when throttling is accompanied by a fall of engine speed, as in level flight, the L.M.E.P. does not increase but in this supercharged engine it decreased, and by slightly more at high altitudes than at ground-level. Some further observations on two 9-cylinder air-cooled radial engines,[47, 60] normally aspirated, are shown in fig. 132, where the tests were made only at the speeds and throttle settings

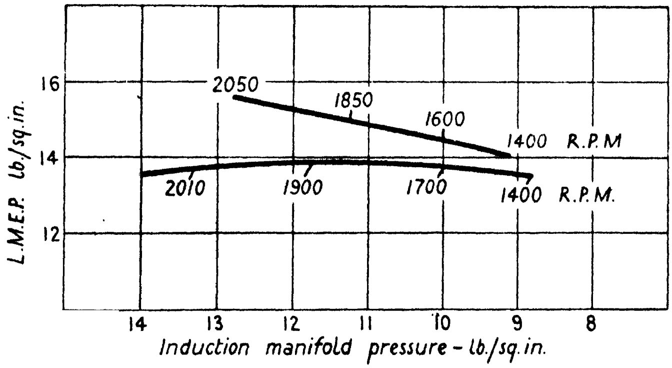

FIG. 132. Observed variation of L.M.E.P. in a 9-cylinder air-cooled engine. Bore and stroke 5¾ in. × 7½ in. Normal full speed 2,000 r.p.m. Maximum permissible speed 2,200. Throttle position and r.p.m. related as in level flight near the ground.

corresponding to level flight near the ground. In one engine the loss was sensibly constant and in the other it showed, as in the engine of fig. 131, a slight fall as the throttle was closed; but this was because, owing to special experimental arrangements at the air intake, there was a depression of 2 lb. per sq. in. in the induction manifold even when on full throttle. The result is an increase of L.M.E.P. of about the same amount in the full-speed test. With allowance for this special circumstance the two engines check one another in showing an L.M.E.P. of $14 \pm \frac{1}{2}$ lb. per sq. in. throughout the speed range for level flight near the ground. All these figures, it should be observed, are only from motoring tests. How far they represent the true losses when the engine is running under power has yet to be considered.

ART. 73. *Analysis of the L.H.P. Mechanical friction and pumping loss.*

If one imagines an engine being motored round in a perfect vacuum, all metal and oil temperatures being normal, then the power required would be very nearly equal to the mechanical friction

loss under real conditions. Friction under the imagined conditions would be slightly reduced because of there being no gas forces on the pistons, but in a motoring test the mechanical friction is nearly all due to shearing of the oil film and independent of the gas forces. To a first approximation, therefore, the mechanical friction loss is a function only of the speed of revolution N, provided the oil viscosity is not affected by any changes of temperature during the test.

When gas has to be drawn in and expelled every cycle there is in addition the pumping loss, which is derived from restriction of the air-flow either at the throttle or the valves. Even if there were no restriction to flow at the valves there would be a pumping loss in a throttled engine, and in this ideal case the total lost power at a given speed and air temperature, expressed as L.M.E.P., would be simply

$$(\text{L.M.E.P.}) = (P-P_m)+F, \tag{68}$$

in which P represents the atmospheric pressure (at any altitude), P_m is the pressure in the induction manifold, and F represents the mechanical friction and is a function only of the speed N and the oil viscosity μ.

In fact, of course, there is some restriction both at the inlet and exhaust valves, which will depend upon the pressure and density of the air, the rotation rate, and the design of the engine (valve timing port area, etc.). It can be shown by dimensional analysis that the pressure differences due to restrictions at the valves and throttle must be proportional to a function of the form

$$Pf\left(\frac{N}{\sqrt{T}}\right),$$

in which T is the absolute temperature of the air of pressure P, and $f(N/\sqrt{T})$ represents some function of $N/\sqrt{T}$. N is the rotation rate of the engine. The form of the function of $N/\sqrt{T}$ that should multiply P can only be settled by examining the results of experiments, and this will be done presently. Leaving it undetermined for the moment we may write, as an expression for the L.M.E.P. which allows for changes of speed and air temperature both at the valves and throttle,

$$(\text{L.M.E.P.}) = Pf\left(\frac{N}{\sqrt{T}}\right)+P_m f\left(\frac{N}{\sqrt{T}}\right)+f'(\mu N), \tag{69}$$

in which P is the atmospheric pressure, P_m is the pressure in the induction manifold, and $f'(\mu N)$ represents the mechanical friction loss. As regards the oil viscosity, μ, in this last term, we can only assume it to remain constant during a test or series of tests, although it cannot, in fact, be very closely controlled. Fig. 128 has

shown that the cylinder-wall temperature has a substantial effect on the L.M.E.P., but it must be remembered that great accuracy is not required. If estimates of the L.M.E.P. are correct to 1 lb. per sq. in. that will suffice, and 1 lb. per sq. in. corresponds to a change of cylinder-wall temperature of 10° C., according to fig. 128.

Assuming, then, that μ is a constant, the mechanical friction becomes a function of N only. The pumping loss may be further analysed so as to show the loss due to the inlet valve and exhaust valve separately, by writing

$$(\text{L.M.E.P.}) = Pf\left(\frac{N}{\sqrt{T}}\right)+P_m f\left(\frac{N}{\sqrt{T_m}}\right)+P_E f\left(\frac{N}{\sqrt{T_E}}\right)+f'(N),^* \quad (70)$$

in which P_m and T_m are the pressure and temperature in the inlet manifold and P_E and T_E those in the exhaust pipe. The exhaust pressure $P_E = P$ the atmospheric pressure, and in a motoring test $T_E = T = T_m$ with sufficient accuracy. An important point to notice here is that the contribution of the exhaust stroke to the L.M.E.P. will be much less when under power than in a motoring test. At ground-level T would be of the order 300° C. abs. and T_E about 1,200° C. abs., and it follows that the exhaust stroke contribution when under power at a certain speed N will be the same as when motoring at half that speed, $N/2$.

From the results of motoring tests of which the results were given in figs. 124 and 125 it is clear that the L.M.E.P. was a linear function of the speed, and it follows therefore that

$$f\left(\frac{N}{\sqrt{T}}\right) = \text{constant} \times \frac{N}{\sqrt{T}}.$$

Furthermore, since engines are designed for different maximum speeds, it will make the analysis more applicable to different engines if all speeds are specified as fractions of the normal full speed of the engine, N_0.

We may then write down an expression for the L.M.E.P. in a motoring test, expressed in atmospheres, as follows,

$$(\text{L.M.E.P.})_{\text{atm}} = \underbrace{\frac{A}{\sqrt{T}}\frac{N}{N_0}P}_{\substack{\text{throttle and}\\ \text{exhaust valve}}} + \underbrace{\frac{B}{\sqrt{T}}\frac{N}{N_0}P_m}_{\text{inlet valve}} + \underbrace{C\frac{N}{N_0}+D}_{\substack{\text{mechanical}\\ \text{friction}}}. \quad (71)$$

contribution from:

* There is the assumption here that the form of the function of $N/\sqrt{T}$ is the same in the first three terms on the right-hand side of the equation. Strictly speaking, this is doubtless incorrect; but within the required ranges of variation of N and T the results show that it is sufficiently correct for all practical purposes.

The constants A, B, C, and D are determinable from the observed variation of the L.M.E.P. with speed, atmospheric pressure, and degree of throttling, as illustrated in figs. 124–6 and 130–1 of the last article.

Choosing that one of the engines in fig. 126 for which the results of throttled tests are given in fig. 130, namely engine 5, the linear

TABLE 48

Variation of L.M.E.P. with throttle position, atmospheric pressure, and speed for engine no. 5 in figs. 124, 126, *and* 130. *Air temp.* $T = 288$ *deg. abs.*

Fixed conditions		*L.M.E.P. lb. per sq. in.*
	Variable throttle	
Ground-level	$P_m = 0{\cdot}95 P_0$	16
1,600 r.p.m.	$P_m = 0{\cdot}41 P_0$	20
	Variable atm. pressure	
Full throttle	$P = P_0$	16
1,600 r.p.m.	$P = 0{\cdot}40 P_0$	11·5
	Variable speed	
Ground-level	$N = 1{,}400$	14
Full throttle	$N = 2{,}000$	20

variations of the L.M.E.P. with throttle position, atmospheric pressure, and speed may be summarized as in table 48. Substitution in the formula gives the following values for the constants

$$A = 21$$
$$B = -10{\cdot}6$$
$$C = 0{\cdot}72$$
$$D = 0.$$

The final equation for the L.M.E.P. under any conditions, according to the experimental results on the engine chosen is, therefore,

$$(\text{L.M.E.P.})_{\text{atm}} = \frac{21}{\sqrt{T}}\frac{N}{N_0}P - \frac{10{\cdot}6}{\sqrt{T}}\frac{N}{N_0}P_m + 0{\cdot}72\frac{N}{N_0}. \qquad (72)$$

It is of interest to obtain from the formula the relative magnitude of the pumping and mechanical loss under some particular conditions. At full throttle and full speed, for example, the pumping loss will be found to be rather less than half of the total, at ground-level; while in level flight under throttled conditions the pumping loss may become about 25 per cent. greater than that due to mechanical friction.

Even at full throttle there will normally be a depression in the inlet manifold of an engine at full speed equal to about 1·5 in. of mercury at ground-level. This means that in such a test P_m would have the value $0{\cdot}95P$, instead of being simply equal to P. In determining the constants of the equation from the data available it has been assumed that $P_m = 0{\cdot}95P$ in all full-throttle tests.

Although determined from experimental results on a single engine, an equation such as that given above will be found to hold good for other engines of the same type, and can quickly be modified so as to give the variation of the L.M.E.P. of other types even when nothing more than a single observation of the L.M.E.P. at full throttle and at some one speed has been made. Thus, in the case of the 9-cylinder radial engine, for which some results were given in fig. 132, the L.M.E.P. at full throttle and at the speed of 2,010 r.p.m. was 13·7 lb. per sq. in. and N_0 for that engine was 2,200. Assuming that the reduced loss, as compared with the engine so far considered, may be divided equally between pumping and mechanical friction, then the equation for the L.M.E.P. under any conditions may be written down as

$$(\text{L.M.E.P.})_{\text{atm}} = \frac{15{\cdot}5}{\sqrt{T}}\frac{N}{N_0}P - \frac{7{\cdot}9}{\sqrt{T}}\frac{N}{N_0}P_m + 0{\cdot}535\frac{N}{N_0}. \qquad (73)$$

Now when throttled under the conditions of level flight near the ground until the speed was 1,700 r.p.m., the induction manifold depression on that engine was found to be 9·62 in. and P_m therefore 20·3 in. or 10·0 lb. per sq. in. Taking T as 288 and $N/N_0 = 0{\cdot}77$, the formula gives the total L.M.E.P. as 13 lb. per sq. in. under those conditions, and, of that, about 7 lb. per sq. in. as due to the pumping loss. This is within about $\frac{1}{2}$ lb. per sq. in. of the measured value given in fig. 132, and variations of that amount or more must be expected in an air-cooled engine on account of variations of cylinder temperature, and hence of the mechanical friction. It will be remembered that according to fig. 128 a change of jacket-water temperature of 10° C. would increase or decrease the L.M.E.P. by 1 lb. per sq. in. All the observations on water-cooled engines illustrated in figs. 124 and 126 were made with the same jacket outlet temperature of 70° C.

No check has so far been made of the effect of inlet air temperature upon the L.M.E.P., and the experimental data for doing so is rather scanty. That given for a 12-cylinder water-cooled engine in fig. 127 shows that at full speed the L.M.E.P. was 20·3 lb. per sq. in. when $T = 288$ abs. and was reduced to 19·15 at $T = 353$ abs. Equation (72) gives L.M.E.P. values of 20 and 19·1 for full speed and the

same temperature conditions, so that in the trials of fig. 127 the variation of the pumping loss was exactly in proportion to $1/\sqrt{T}$. The data from a single engine, given in fig. 127, are all that are available on the effect of inlet air temperature with other conditions constant, except for some rather scanty observations in the same report on an air-cooled radial engine in which no consistent variation of L.M.E.P. with temperature could be found. On the other hand, the exceptionally small reduction of the L.M.E.P. with pressure shown in fig. 126 for two tests on a Liberty engine, in which the inlet air temperature was lowered with the pressure, cannot all be put down to the lowering of the air temperature if the effect is proportional to $1/\sqrt{T}$. This would only account for about half the difference between these and the other tests. An inlet air temperature below 0° C., however, in a motored engine, even with the jackets maintained at 70° C., might easily give a large increase of mechanical friction due to an increase of oil viscosity on the pistons, and this probably accounts for the figures in the Liberty engine tests which were given in detail in tables 46 and 47. The true effect of air temperature must certainly be small, and a variation proportional to $1/\sqrt{T}$, which is theoretically consistent with the observed linear effect of a change of speed, may probably be accepted for all engines without serious error.

There remains to be considered the effect of the exhaust gas temperature when running under power. This temperature being about four times that when motoring, the contribution of the exhaust valve to the L.M.E.P. should be halved. Allowance might be made in the equation for the L.M.E.P. by retaining the separate term representing the restriction at the exhaust valve in equation (70); but it would only be possible to determine the necessary constants if light spring diagrams were taken, so that the separate contributions of the inlet and exhaust valves to the pumping loss in the motoring test could be calculated. Failing that, one may guess the contribution of the inlet valve to be rather more than half the total, with normal valve timing and gas speeds. The formula gives, for the air-cooled radial engine of fig. 132 at full throttle and full speed, an L.M.E.P. of 14·8 lb. per sq. in. of which 6·9 is pumping loss. Guessing rather more than half of this,* say 4 lb. per sq. in. to be due to the exhaust valve when motoring, then by reason of the high temperature of the gases when working under power this would be reduced to about 2 lb. per sq. in. The effect would be less at heights, in proportion to the pressure.

* Mean gas speeds are generally higher through the exhaust than through the inlet valves.

Hitherto it has been customary to assume that this undoubted reduction in the pumping loss, demonstrable by taking light spring indicator diagrams, is compensated by an equal increase in the mechanical friction due to the gas forces during the working stroke; with the result that the L.M.E.P. as found from a motoring test still holds good when running under power. It is possible that this assumption involves an over-estimate of the L.M.E.P. when under power, and that the value obtained by taking the exhaust valve contribution as half that in a motoring test would be nearer the truth. But the calculations of the rates of fuel consumption at different altitudes, to be given in Chapter XIV, have been based on the usual assumption as to the lost power under working conditions, and they show a satisfactory agreement with fuel consumption measurements made in flight.

It should be mentioned that in any motoring test it is very important to be on the look out for any resonance between the engine speed and air-flow in the exhaust system, for synchronism of this kind can make a difference of 2 or 3 lb. per sq. in. in the pressure at the exhaust valve; and similarly a supercharging effect can be obtained on the inlet side if each cylinder is fed through a separate induction pipe.

XIII

ALTITUDE AND POWER OUTPUT

ART. 74. *The normally aspirated engine. Power estimation from ground-level data.*

For practical purposes in the estimation of aeroplane performance what is required to be known about an aero-engine is some factor, ϕ, with which to multiply the ground-level B.H.P. in order to obtain the shaft h.p. available at any height where the pressure, density, and temperature of the air relative to ground-level conditions are p, σ, and θ. In arriving at such a power factor it is convenient to work via the conception of I.H.P. and to obtain first the corresponding factor ψ which defines the I.H.P. in terms of its ground-level value. It has been shown that there is a great deal of experimental evidence that goes to prove that the I.H.P. of any normally aspirated engine, with its ignition timing and fuel-air mixture properly adjusted, is always proportional to the weight of fresh charge per cycle. It is possible, therefore, to derive a power factor ψ for multiplying the I.H.P. at ground-level which is independent of any particular engine. When ψ has been defined in terms of p and θ for all engines, it then remains only to obtain a relation between ϕ and ψ which can be determined for any particular type of engine by a ground-level trial.

It was shown in art. 19 that the I.H.P. of a single-cylinder supercharged engine varied almost exactly with the weight of fresh charge per cycle, and this result has been found to hold good also in trials of multi-cylinder normally aspirated engines in which the external air temperature was varied between $-20°$ and $+40°$ C. and the pressure from normal atmospheric down to $p = 0{\cdot}4$, corresponding to a height of 24,000 ft. in the standard atmosphere.[57, 58] At extreme heights and low temperatures irregularities of fuel evaporation and mixture burning no doubt upset the normal functioning of the engine, but so long as no special factors come in to alter its thermal efficiency there is no reason why an engine should not always produce the same number of ft. lb. of indicated work from each lb. of air drawn into its cylinders.

We may assume that the pressure in the cylinder when the inlet valve closes will be always very nearly atmospheric, and if the temperature of the mixture were proportional to that of the outside air, then the I.H.P. would fall off with altitude in proportion to σ, and we should have, simply,

$$\psi = \sigma.$$

It was shown in arts. 69 and 70 that the mixture temperature in the cylinder varies less rapidly than that of the outside air. In the extreme case we might imagine that as an aeroplane rises into colder and less dense air sufficient heat can always be picked up by the in-going mixture from the induction pipes, valves, and cylinder walls to produce the same final temperature before compression under all conditions. The I.H.P. would then be independent of the external air temperature, and we should have

$$\psi = p.$$

Flight trials have never indicated that the B.H.P. of an engine is wholly independent of the external air temperature. The degree to which the influence of this is felt probably varies to some extent in different types of engine, and it appears to vary also with the height. Above about 6,000 ft. on an average day the air temperature affects the power of a normally aspirated engine very little, and the engine behaves very nearly as though its power were a function of the air pressure only. At lower heights, however, observations indicate that the air temperature has a noticeable influence, and the power variation veers towards a dependence on air density. It may well be that, as the height increases and the temperature falls, there comes a point beyond which the fuel is less and less completely evaporated before the inlet valve has closed; and this failure to evaporate all the fuel would compensate for the falling external temperature and result in a tendency towards a constant gas temperature at the moment of closure, with a consequent power variation depending only on the external pressure.

The use of the terms 'pressure basis' and 'density basis' for the calculation of aircraft performance, by which is meant an assumption that the engine power is a function of pressure *only* or of density *only*, has almost suggested that pressure and density are two separate factors which influence engine power; and that the use of one of them in performance calculations might be shown to be correct, to the exclusion of the other. From what has been written above, and in art. 69, it will be obvious that engine power is necessarily a function of the outside pressure, and the degree to which the external air temperature affects the conditions inside the cylinders determines to what extent the power is also a function of the surrounding air density.

The data given in art. 70 showed that the I.H.P. of an engine varied very nearly in proportion to $1/\sqrt{T}$, with indications of a more rapid variation at the highest temperature. The power factor for I.H.P., ψ, may therefore be expressed in the general form

$$\psi = p\theta^{-a} \tag{74}$$

in which a is a number less than 1, and is very closely $\frac{1}{2}$ for normal ranges of air temperature.

Under all circumstances

$$\theta = \frac{p}{\sigma}$$

and therefore ψ may be expressed as

$$\psi = p^{1-a}\sigma^{a} \tag{75}$$

in which the two conditions $a = 1$ and $a = 0$ represent the extremes in which the cylinder-gas temperatures are (1) proportional to, and (2) independent of, the external air temperatures. The bench tests quoted in art. 70 have shown that over the range of conditions normally met with $a = \frac{1}{2}$ and hence

$$\psi = p^{\frac{1}{2}}\sigma^{\frac{1}{2}}. \tag{76}$$

Turning now to the power factor ϕ, for the B.H.P., it was mentioned in art. 69 that if this were assumed to be $\sigma\theta^{\frac{1}{2}}$, which is the same as $p\theta^{-\frac{1}{2}}$ or $p^{\frac{1}{2}}\sigma^{\frac{1}{2}}$, the result would be to reduce the over-estimate of the engine power involved in assuming it proportional simply to σ by about 4 per cent. at 10,000 ft. A more accurate expression for ϕ may now be derived on the basis of the known variation of L.H.P. as given in art. 71. It was there shown that the pumping loss was subject to the same law of variation as the I.H.P., namely that it varied as $P/\sqrt{T}$ for a given rotation rate, and that equation (71), based on the assumption that the mechanical friction was a function of the speed only, gave consistent and properly representative figures. There is no reason to expect that in flight the mechanical friction will not be always very nearly the same at the same r.p.m. whatever the height, for against any slight drop due to a reduction of the gas forces in a normally aspirated engine may be set the likelihood of a slight increase of oil viscosity in the cylinders, owing to the lower inlet air temperature.

If I_0 represents the I.H.P. at ground-level and m the mechanical efficiency at normal speed and full throttle, as determined by bench tests, then the L.H.P. at ground-level is $I_0(1-m)$ and the constant friction loss may be represented by

$$\lambda I_0(1-m).$$

For an average aero-engine λ would be about 0·55 according to the figures given in art. 73. With the constants in equation (73) it is 0·53.

It can be shown* quite simply that the relation between ψ and ϕ is then

$$\phi = \psi\left(1+\frac{\lambda-\lambda m}{m}\right)-\frac{\lambda-\lambda m}{m} \tag{77}$$

$$= p\theta^{-\frac{1}{2}}\left(1+\frac{\lambda-\lambda m}{m}\right)-\frac{\lambda-\lambda m}{m} \tag{78}$$

when allowance is made for reduction of the pumping loss in proportion to $p/\sqrt{\theta}$, combined with constant mechanical friction at any given speed of rotation. The effect of making more accurate allowance for the L.H.P. is to lower the B.H.P. at normal r.p.m. by 2·5 per cent. at 10,000 ft. as compared with what it would be if ϕ were assumed equal to $p/\sqrt{\theta}$. This is when $\lambda = 0{\cdot}55$ and $m = 0{\cdot}9$. If the mechanical efficiency were 85 instead of 90 per cent. the formula would lower the estimated B.H.P. at 10,000 ft. by 4 per cent. instead of 2·5. An inaccuracy in the estimate of λ, however, is unimportant, since a change from 0·5 to 0·6 makes only 0·5 per cent. difference in the estimated B.H.P.

The calculated value of ϕ for any height can only be applied to find the *power* at the normal r.p.m., but after calculating the I.M.E.P. and B.M.E.P. at the normal r.p.m. the L.M.E.P. for any other speed may be taken as in proportion to the speed, according to equation (71). A comparison of the formula with some experimental results is to be given in the next article.

ART. 75. *Power measurements in flight.*

The torque actually produced by an engine at a given speed when in flight can be directly measured by some form of dynamometer incorporated in the airscrew hub, if that is available, or can be deduced from the performance of the aeroplane while using the airscrew as a dynamometer, after previous calibration on the ground. The second method gives the apparent power variation after this may have been affected by slight distortion of the airscrew under its working load, which affects its calibration, and by other factors. As a check upon the engine, therefore, the first method, given a satisfactory instrument, is likely to be the more accurate. The production of a reliable instrument for measuring engine torque in the air is a matter of much difficulty, but with very special precautions it is possible to obtain consistent readings, as will be seen from curve *A* in fig. 134 where the individual observations are plotted.[63] A further check upon engine power in the air, of an indirect nature, can be obtained by observing the rate of air consumption,[64] and then

* For proof see Appendix III, p. 389.

calculating the power output from the correlation between power and rate of air consumption as determined by bench tests. Some figures from an experiment of this kind are also included in what follows.

Of the three curves in fig. 133 the uppermost shows how the power of an engine would vary with height in the standard atmosphere if it were proportional to σ, the second if proportional to $p\theta^{-\frac{1}{2}}$ (which is closely correct for the I.H.P. but not for the B.H.P.), and the third curve is based on equation (78). It will be seen that the

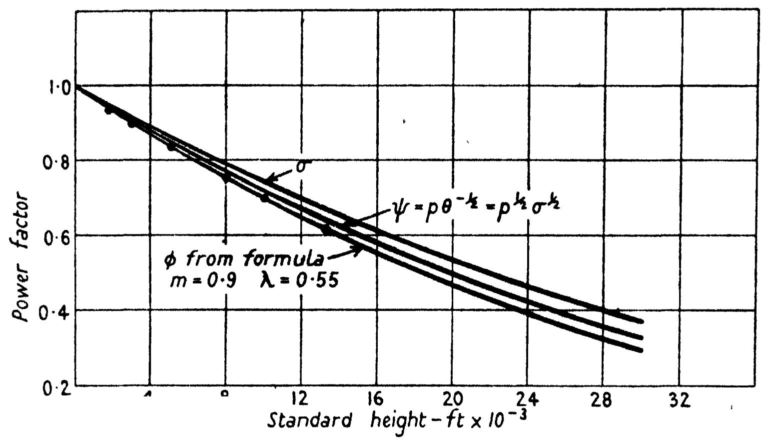

FIG. 133. Variation of engine power with height in the standard atmosphere.

power factor $p\theta^{-\frac{1}{2}}$ or $\sigma\theta^{\frac{1}{2}}$ when applied to the B.H.P. gives an estimate of power greater than the formula by 3 per cent. at 10,000 and 5 per cent. at 20,000 ft.

In fig. 134 are three series of tests showing the observed falling off of power with height at constant speed, the first two series made with the Bendemann hub dynamometer[63] and the third using a calibrated airscrew for measuring the power. In curves *B* and *C* the observations are rather scattered, and it must be remembered that even the consistent observations of curve *A* may be subject to a systematic error of the instrument due, for example, to changes of oil viscosity with air temperature. The curves are given for comparison with the power variation defined by equation (78); but in view of the check given in the next paragraph, summarized in table 49, the values obtainable from the formula are considered to be the more accurate.

In *R. and M.* 1141 are given values for the power factor ϕ deduced from the type trial reports of large numbers of aeroplanes, both with air-cooled and water-cooled engines. The air-cooled and water-

cooled engines show a difference of power factor amounting to just under 3 per cent. at 25,000 ft. in favour of the air-cooled. This is of particular interest in connexion with equation (78), because a difference of mechanical efficiency of 2 per cent. at ground-level would, according to the formula, account for this difference of power factor, and such a margin of mechanical efficiency is to be expected, in favour of the air-cooled radial engine, by reason of its short crankshaft and high cylinder temperatures.

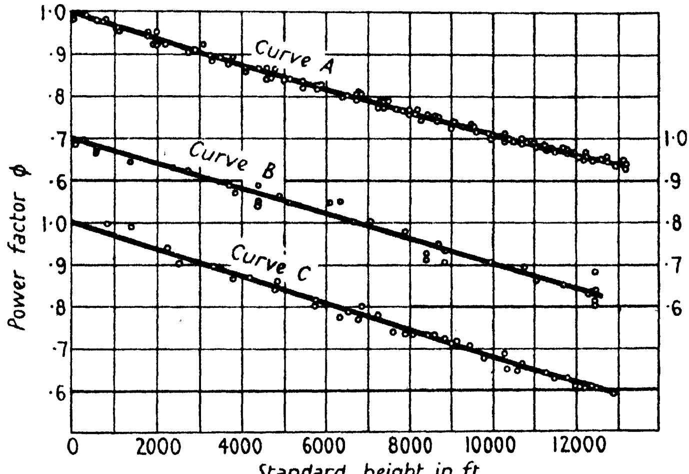

FIG. 134. Observed values of the power factor ϕ, for B.H.P., against height.

The numerical differences being too small to show satisfactorily in the form of a diagram, the comparison between the values of the power factor given by equation (78) and those obtained from type trials is given in table 49. The mechanical efficiency for the different

TABLE 49

Comparison of the power factor ϕ as obtained from the formula of art. 74, with the mean values deduced from type trials of large numbers of aeroplanes.

Standard height	*ϕ deduced from type trials*			*ϕ from formula, eq. (78).*
	Air-cooled engines	*Water-cooled engines*	*Mean of all engines*	$m = 0·9$ $\lambda = 0·55$
0	1	1	1	1
5,000 ft.	0·84	0·838	0·838	0·835
10,000 ft.	0·695	0·691	0·693	0·695
15,000 ft.	0·569	0·563	0·567	0·572
20,000 ft.	0·456	0·447	0·454	0·47
25,000 ft.	0·358	0·348	0·355	0·37

engines not being available the formula can only give a mean value, and as before m and λ have been taken as 0·9 and 0·55, which are appropriate to a radial air-cooled engine. The discrepancy is less than 1 per cent. up to 15,000 ft. Above this the observations show a slightly greater fall of power than that given by the formula, as would, indeed, be expected where carburation is likely to be imperfect.

A series of trials,[64] in which the rate of air consumption was measured in flight up to heights of rather over 13,000 ft. by means of the Callendar air-flow meter, are of special interest, since they give an independent check upon the methods given in the last article for estimating engine power, and also for the other essential data which can be gleaned from them. The engine was the 6-cylinder Puma, water-cooled, of bore and stroke 5·7 in. × 7·48 in. (145 mm. × 190 mm.). Bench tests were first made from which the correlation between power and air consumption was obtained, both at full throttle for a range of speeds, and also under throttled conditions. Throughout the full-throttle series, from 1,100 to 1,500 r.p.m., the I.H.P. was found to be always very closely 1·200 × the air consumed in gm. per sec.

Using the observed rates of air consumption at different heights, as given in col. 2 of table 50, a series of values of ψ can be calculated

TABLE 50

Air consumption at different heights and the values of the I.H.P. and of the power factor ϕ deduced therefrom for a Puma engine. Bore and stroke 5·7 in. × 7·48 in.

Speed about 1,400 r.p.m.
Mechanical efficiency at that speed at ground-level = 0·89
I.H.P. = 1·200 × air consumption in gm. per sec.

Height	*Air consumption observed gm. per sec.*	*I.H.P. from bench test correlation*	ψ	ϕ
ground-level	220	264	1	1
1,800	207	248·5	0·94	0·936
3,000	195·5	234·5	0·905	0·898
5,100	188	226	0·845	0·834
8,000	171	205	0·765	0·749
10,000	157	188·5	0·714	0·694
13,300	139·5	167	0·638	0·613

which give the observed fall of I.H.P. as deduced from the rate of air consumption. The mechanical efficiency of the engine at 1,400 r.p.m. was also observed, as 0·89, and using this value for m and

taking $\lambda = 0{\cdot}55$ as before, values of the power factor ϕ for the B.H.P. have been found, as in the last column of the table. The values of ψ have been taken from a smooth curve, but the greatest variation of any observation from the curve was only just over 1 per cent.

The values of ϕ so found have been plotted as the dots on the lowest curve in fig. 133. The curve itself was drawn from equation (78); i.e. it assumes that $\psi = p\theta^{-\frac{1}{2}}$, and the almost perfect agreement of ϕ as derived from these air measurements is therefore a satisfactory confirmation that the I.H.P. follows that law of variation.

Valuable data about the volumetric efficiency of the Puma engine were also obtained in these trials, which enable the rise of temperature of the air as it enters the cylinder to be calculated. The temperature and pressure of the air in the induction manifold was measured at each height, and in col. 2 of table 51 are given the calculated volumetric efficiencies at 1,400 r.p.m., expressed as the actual mass of

TABLE 51

Volumetric efficiencies and temperature rise of the in-going air at different heights. Puma engine at 1,400 *r.p.m.*

Height	*Volumetric efficiency per cent.*	*Temperature rise of in-going air. °C.*
2,000	82·4	
3,000		60
4,000	82·5	
5,100		59
6,000	82·9	
8,000	83·3	55
10,000	83·7	50
12,000	84·2	
13,300		49
14,000	84·7	

air drawn in per cycle divided by the mass required to fill the stroke volume at the pressure and temperature of the manifold. One may assume that as the inlet valve closes the pressure in the cylinder is the same as that in the manifold, and that the volumetric efficiency is therefore the ratio of the absolute temperatures of the charge before and after entering the cylinder, but before mixing with the residual exhaust gas. Col. 3 of the table shows the rise in temperature of the in-going air, the drop from 60° to 49° C. between 3,000 and 13,000 ft. being a reflection of the cooler condition of the pistons and valves owing to a drop in the power output from about 220 to 150 B.H.P. The jacket-water temperature was maintained nearly constant throughout. It should be mentioned that above 5,000 ft. the air-petrol ratio was kept between 13·2 and 13·6, and that below

this it was richer, down to 10·9, so that the observed increase of volumetric efficiency was not due to extra cooling by fuel evaporation, but to a reduction by the heat picked up by the in-going air from the metal surfaces.

ART. 76. *The correction of engine power to conditions in the standard atmosphere.*

The data from flight tests given in the last article have confirmed the result, derived in the first place from bench tests, that the B.H.P. of an engine at any given speed (to which the B.M.E.P. is proportional) will be given by ϕ(B.H.P.)$_0$, where (B.H.P.)$_0$ is the ground-level B.H.P. and

$$\phi = p\theta^{-\frac{1}{2}}\left[1+\frac{\lambda-\lambda m}{m}\right]-\frac{\lambda-\lambda m}{m}.$$

Since p, σ, and θ, for any height in the standard atmosphere, are related so that $\theta = p^{0\cdot19}$ and $p = \sigma^{1\cdot235}$, the power factor ϕ in the standard atmosphere can be expressed in terms of the pressure only as

$$\phi = p^{0\cdot905}\left[1+\frac{\lambda-\lambda m}{m}\right]-\frac{\lambda-\lambda m}{m} \tag{79}$$

or in terms of the density only as

$$\phi = \sigma^{1\cdot12}\left[1+\frac{\lambda-\lambda m}{m}\right]-\frac{\lambda-\lambda m}{m}. \tag{80}$$

An equation in some such form, showing the variation of ϕ in the standard atmosphere, is all that is required for the estimation of aeroplane performance, which would always be made, of course, in relation to heights in the standard atmosphere. On the other hand, when the problem is that of reducing the results of flight tests obtained on a non-standard day to what they would have been in the standard atmosphere, then the variation of B.H.P. with height does not provide what is required. The essential thing for this purpose is to know as accurately as possible to what degree the outside air temperature affects the B.H.P. actually available or, in other words, to what degree the effective B.H.P. is to be regarded as a function of the air density; and, for reaching a conclusion on the point, the only satisfactory evidence must be that derived from flight tests. Absolute measurements of power are not required, but only the effect of air temperature upon the engine power at full throttle at various heights defined by the pressure.

We have seen that in general the power factor ψ for the I.H.P. can be expressed in the form

$$\psi = p^{1-a}\sigma^{a},$$

and it might be expected that the power factor for the B.H.P. would vary so as to be expressible as some function of ψ. If the L.H.P. and L.M.E.P. were functions of the speed only, i.e. if the losses were not dependent on atmospheric conditions, then it can be shown[65] that the B.H.P. would be expressible in the form

$$\phi = \psi^{1+e} = (p^{1-a}\sigma^{a})^{1+e}.$$

It is quite clear, however, from the results given in art. 71 that the pumping loss, and therefore the L.M.E.P. as a whole, *does* depend on the atmospheric conditions, and therefore that ϕ cannot satisfactorily be expressed simply as some power of ψ. This, however, does not matter for present purposes, and we can nevertheless write

$$\phi = f(p^{1-a}\sigma^{a}),$$

meaning that whenever the I.H.P. has a definite value, so also has the B.H.P. For the correcting of flight trials to conditions in the standard atmosphere we do not need to know what is the form of the function represented by f, but only the value of a in the factor within the bracket. The value of a is what defines the 'basis' upon which the reduction to standard conditions is to be made, and the method by which the best value to adopt can be settled from flight trials must now be explained. The final conclusion *may*, and in fact does, agree with the bench tests, but this is not a thing to take into account at present. The sole criterion for the correct value of a, as applied to any particular aeroplane and engine, is that if the aeroplane had been tested on two days on which there was a wide difference between the atmospheric temperatures, then the calculated performance in the standard atmosphere based upon either day's results would be the same.

During routine performance testing it is not often that between separate tests on the same machine the difference of air temperature is sufficient to prevent the true effect of a change of temperature from being masked by the random errors of measurement. In seeking the necessary data, therefore, a method has to be adopted of selecting from a large number of performance trials those in which results are available on the same machine and engine, taken on days of as widely differing temperature as possible. This has been done, and the conclusions from such a statistical examination are presented in *R. and M.* 1532 after subjecting the data to a careful system of 'weighting' whereby reliance is placed upon individual results not only in accordance with the consistency between separate observations and with the number of separate trials carried out, but according to the differences of temperature which existed between the occasions of the various trials. As a result of examining the type trials of

seventeen aeroplanes with supercharged engines, and a large number with normally aspirated ones, the mean values found for a were 0·48 for supercharged engines and 0·43 for the others. If a is taken as 0·5 for all engines, then the reduction to standard conditions will be quite sufficiently accurate for all practical purposes and the basis of reduction agrees with the form of the power factor found from bench tests, namely

$$\phi = f(p^{\frac{1}{2}}\sigma^{\frac{1}{2}}).$$

The idiosyncrasies of individual engines, or uncertainties of measurement, were sufficient in some cases to make the apparently correct value of a as far from the mean as to correspond to a pure pressure basis ($a = 0$) and in others to a pure density basis ($a = 1$); but the nearness of the general mean to the bench-test results is sufficient to justify the adoption of $a = 0{\cdot}5$ for normal use.

The method of deducing the most appropriate value of a for any particular aeroplane from the observed effect of a change in atmospheric temperature upon the rate of climb and engine r.p.m., or rather upon $V_c\sqrt{\sigma}$ and $N\sqrt{\sigma}$, at different heights, has been given in *R. and M.* 1316. It rests upon a perfectly general proposition, also proved in the same report, that if the full-throttle engine power Z in an atmosphere of pressure P and temperature T can be expressed in the form

$$Z = f(PT)Z_0 \tag{81}$$

and if the ground-level power Z_0 can be expressed in the form

$$Z_0 = AN^r,^* \tag{82}$$

then at any given value of the function $f(PT)\sigma^{(1-r)/2}$ the fixing of the value of any one of the four functions $V\sqrt{\sigma}$ (or V_i), V/ND, $V_c\sqrt{\sigma}$, and $N\sqrt{\sigma}$, fixes all the rest. The special condition of maximum level speed would make $V_c = 0$, and either in these circumstances or when V_c is a maximum, the proposition means that V_i, V/ND, and $N\sqrt{\sigma}$ must depend only on the value of the function $f(PT)\sigma^{(1-r)/2}$.

If it were possible to plot these three 'performance factors' against values of the function, then the same curves would always be obtained no matter how the relation between P and T might have varied between different days on which the observations were made. Although the form of the complete function is not known, we have seen that the power factor for the B.H.P. of the engine, which may be identified with the first part, $f(PT)$, may also be expressed as some function of $(p^{1-a}\sigma^a)$, and that in this expression a defines the basis upon which reduction to standard conditions is to be made.

* As a rule the B.M.E.P. of an engine falls very slowly with speed until its full speed is approached (see fig. 23) and the power is therefore nearly proportional to N. The index r is consequently close to unity under climbing conditions.

If the value of a were known it would be possible to plot the performance functions against values of $(p^{1-a}\sigma^{a})$ at a series of heights defined by the pressure, and further, if r is nearly unity so also is $\sigma^{(1-r)/2}$, so that plotting against $(p^{1-a}\sigma^{a})$ is the same thing as plotting against the complete function $f(PT)\sigma^{(1-r)/2}$. So long as r may be taken as equal to unity, therefore (see note, p. 350), and if a is known, it is possible to calculate the performance of an aeroplane in one atmosphere from the known performance in another, for all that would be necessary would be to plot V_i, $V_c\sqrt{\sigma}$, or $N\sqrt{\sigma}$ against $(p^{1-a}\sigma^{a})$, then plot two height scales against $(p^{1-a}\sigma^{a})$ using the values of p and σ at various heights in the two atmospheres. By comparing the curves of the performance quantities with the curves of heights, all plotted against $(p^{1-a}\sigma^{a})$, the step from one atmosphere to another is obtained.

By the converse of this process, when the performance has been found on two different days, neither of them standard, it is possible to find the necessary value of a to give the same performance in the standard atmosphere. The process is described in *R. and M.* 1532 and need not be elaborated here further than to explain that a trial and error method is employed. The tests covering a range of temperature are first reduced to standard conditions on the assumption that $a = 0$ (pressure basis), and a comparison of the results with those of the other extreme assumption $a = 1$ (density basis) will usually suffice to settle the intermediate value of a which would bring the performances obtained on days of different temperature into coincidence when reduced to the standard atmosphere.

For such an examination into the proper basis of reduction there is no need to make use of more than one of the three quantities V_i, $V_c\sqrt{\sigma}$, and $N\sqrt{\sigma}$ for comparing the performances on different days. In *R. and M.* 1532 the mean value of a was deduced by examination of both V_c and $N\sqrt{\sigma}$ at different heights, and the mean value from each set of observations was not quite the same. The conclusion was reached that the results based upon observations of $N\sqrt{\sigma}$ have a greater reliability than those from $V_c\sqrt{\sigma}$, probably because the former are unaffected by casual up-and-down air currents in the atmosphere in the way the apparent rates of climb would be.

The only assumptions involved in settling the question of the basis of reduction, as defined by the index a, are that the B.H.P. at ground-level can be represented by $Z_0 = AN$, and that at any other height the power factor ϕ or $f(PT)$ can be expressed as some function of $(p^{1-a}\sigma^{a})$, and that the airscrew characteristics are independent of centrifugal and aerodynamic stresses and of compressibility in the atmosphere. As already mentioned, the method only gives information about the *form* of the law of power variation and does not tell

us what the law is. The information obtained from flight tests concerns, in general, the complete function $f(PT)\sigma^{(1-r)/2}$, and this information cannot be analysed so as to give information about the power factor $f(PT)$ or $\sigma^{(1-r)/2}$ taken separately. It is true that from ground-level bench tests we know that over a limited speed range r is very nearly unity; and so far as this is so the quantity $f(p^{1-a}\sigma^{a})$ found from flight tests to be, on an average, $f(p^{\frac{1}{2}}\sigma^{\frac{1}{2}})$ is in fact the same as the power factor $f(PT)$. This identity of the two, however, is strictly correct only so long as the B.M.E.P. is constant at different rotational speeds; for only then is $r = 1$, so that the B.H.P. $= AN$ while operating at a constant atmospheric pressure and temperature.

Art. 77. *The supercharged engine.*

The most common form of supercharged engine is one in which the supercharger is driven by gearing from the crankshaft and is controlled so as to maintain a constant pressure (often called the boost pressure*) in the induction manifold up to a certain 'rated altitude'; and above that height the air-supply pressure falls off with the atmospheric pressure. The maximum compression ratio of which the supercharger is capable, and therefore the rated altitude, will depend upon the speed of the engine to which it is geared and will therefore be greater at full speed in level flight than when climbing. The accepted rated altitude of an engine is the maximum height in the standard atmosphere to which it can maintain the specified boost pressure at the normal r.p.m., which would correspond rather to climbing than to full-speed conditions.

Constancy of the manifold pressure below the rated altitude must not be supposed to mean a constant power output. Far from it. The B.M.E.P. and the power of the engine at a given r.p.m. will increase at the rate of about 1·1 per cent. per 1,000 ft. up to the rated altitude, in the first place because the mass of air taken by the engine per cycle will increase, at a constant supply pressure, as the outside temperature and pressure fall and as the quantity of residual gas left in the cylinders at atmospheric pressure decreases; and also because of the increase of the balance of positive over negative work during the induction strokes. The exact variation of the B.H.P. available is a result of somewhat complex interactions between the engine and super-

* It has become customary to use the term 'boost pressure' as synonymous with the pressure in the induction manifold, when speaking of supercharged engines. It is an unsatisfactory term, because the designed or 'rated' pressure in the manifold is often just *below* normal atmospheric at ground-level; and to speak of it as a 'boost' pressure at ground-level is a contradiction in terms. Nevertheless the shortness and general acceptance of the term appear to justify its retention.

charger, for a change of the mass-flow of air through the supercharger will alter its efficiency (see fig. 101) and therefore the power absorbed by it per lb. of air delivered to the cylinders. It must be remembered that the pressure in the induction manifold as an aeroplane climbs is controlled by the throttle at the inlet side of the supercharger, which is only opened fully for the first time at the rated altitude. Below that height the quantity of air admitted to the supercharger, or the mass-flow per sec., would be adjusted by the throttle so as to keep the delivery pressure constant. Although the swept volume of the engine would remain constant at constant speed, any increase of volumetric efficiency or lowering of the air temperature would mean an increase of the mass-flow of air through the supercharger, and a change of its efficiency.

It is stated above that the power of an engine at constant r.p.m. would increase by about 1·1 per cent. per 1,000 ft. up to the rated altitude; but this is not what happens when an aeroplane is climbed at a constant boost pressure, and a short digression on the point will serve to bring out some of the interactions of an engine and its supercharger. The aeroplane, climbing at its best climbing speed below the rated height, would maintain a constant A.S.I. reading, and therefore a true speed V in proportion to $1/\sqrt{\sigma}$. Since the A.S.I. reading is constant, so also are the drag of the aeroplane, V/ND, the airscrew efficiency, and the B.M.E.P. of the engine (see Chapter II, p. 38). The constancy of V/ND means that the engine speed during the climb to the rated height is not constant, but increases in proportion to $1/\sqrt{\sigma}$. The effect of this upon the supercharger is to increase the compression ratio and the temperature-rise, the latter nearly in proportion to N^2 and therefore to $1/\sigma$. Since the induction-pipe pressure is constant, the density of the air supplied is reduced; and this lowered density just makes up for the increased volumetric efficiency due to the lower exhaust pressure, with the result that the B.M.E.P. is maintained constant, as stated above.

The fundamental controlling factor upon the boost pressure which the supercharger can be allowed to provide below the full-throttle height is not mechanical strength so much as liability to detonation or, if a non-detonating fuel were available, it would be the difficulty of cooling pistons and valves. These limitations are a serious handicap to the power yielded when taking off by an engine with a gear-driven supercharger. They mean that instead of making full use of the supercharger for increasing the engine power, the air intake must be throttled until the air is delivered at little, if anything, above the atmospheric pressure, although the supercharger, meanwhile, absorbs the full power necessary to drive it when giving

the compression ratio corresponding to the take-off speed of the engine; and, even more serious, the air is delivered to the cylinders at a temperature which may be anything up to 60° C. above the atmospheric temperature, according to the rated altitude for which the supercharger is designed, taking 12,000 ft. as about the practicable limit for a single-stage compressor.

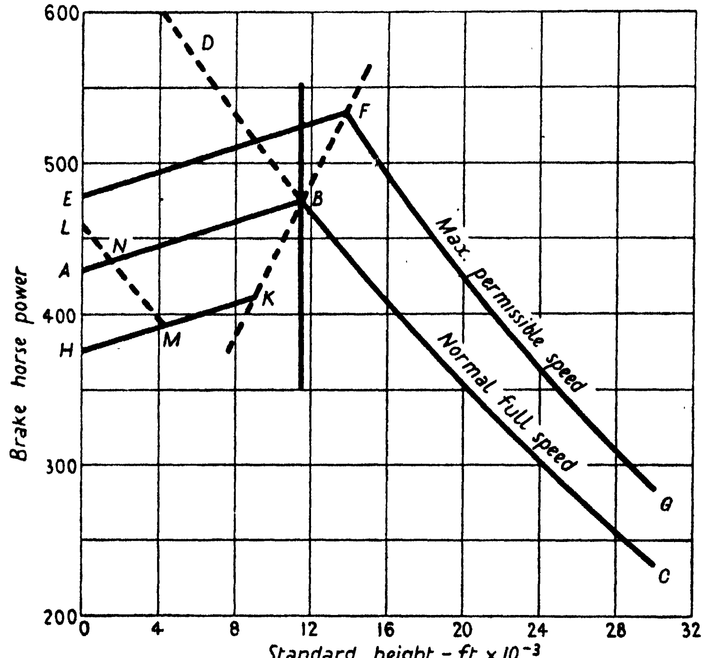

FIG. 135. Typical power curves for a supercharged engine at different heights.

To minimize this handicap it is usual to allow a 'maximum permissible' boost pressure for take-off purposes which is higher than the engine can stand for continuous running, and various forms of control have been devised to safeguard the engine from being overloaded, while at the same time providing as much power as possible for the take-off.

If the control were not of this special kind, but simply maintained a constant boost pressure under all conditions up to the rated altitude, then the form of the power curve at constant speed would be as shown by the curve *ABC* in fig. 135, which is typical for an engine with a rated altitude of 11,500 ft. The upper, dotted portion *BD* does not represent any real performance of the engine, and is put in only to emphasize the effect of the necessary throttling below the

rated altitude. The actual B.H.P. at ground-level was 430, but if it had been possible to allow the boost pressure to rise to what the supercharger would make it at full throttle at ground-level, an engine of the same swept volume would deliver nearly 700 B.H.P.

The portion *AB* of the curve shows the combined effect of the fall of air temperature and the increase of volumetric efficiency when the boost pressure is maintained constant by the gradual opening of the throttle. The B.H.P. at a constant r.p.m. reaches a peak at the rated altitude, for above that the boost pressure falls away nearly in proportion to the external atmospheric pressure: not quite so fast, because the continued fall of air temperature causes a small increase in the compression ratio of the supercharger, at a constant r.p.m.

If *ABC* represents the power at the normal speed of the engine, the peak at *B* is by no means the maximum power it can give. It would probably be the maximum power available at the best rate of climb, because the airscrew is commonly designed to give the normal r.p.m. under those conditions, but at higher forward speeds and engine r.p.m. the maximum power would be increased not only by the increase of speed, but also because the constant boost pressure could then be maintained to a greater height by reason of the higher speed of the supercharger. Thus with the engine for which *ABC* is the power curve at the normal r.p.m. of 2,250, the normal boost pressure of 14·2 lb. per sq. in. can be maintained up to a height of 13,800 ft. when the engine is turning at its maximum permissible r.p.m. of 2,700, as compared with the rated altitude of 11,500 ft. The full-power curve at the maximum permissible r.p.m. would be as shown at *EFG*, and the maximum powers at various speeds and full-throttle heights would lie along the locus *KBF*. The margin between the normal and the maximum speeds is greater than usual in the engine illustrated in fig. 135 (the Rolls Royce Kestrel) but the form of the power curves is similar for all types of engines with geared superchargers. On an average the maximum permissible speed is about 15 per cent. higher than the normal. It is usual to design the airscrew for a supercharged engine so as to allow the revolutions to rise to the maximum permissible when flying level at full throttle at the rated altitude, and this means that below that height the engine torque would only be sufficient to turn the airscrew at the full r.p.m. when the aeroplane was diving. Only under those conditions, therefore, is the portion *EF* of the power curve at full speed possible.

With a fixed pitched airscrew the r.p.m. while taking off would be well below the normal. It was, for example, 1940 in the engine illustrated in fig. 135, and if this speed were maintained in the air

the power curve would be HK, with a peak at 411 h.p. and no more than 376 h.p. at ground-level. In order to improve on these conditions for take-off, a maximum permissible boost pressure of $1\frac{3}{4}$ lb. per sq. in. above atmospheric is allowed, as compared with the normal boost of $-\frac{1}{2}$ lb. per sq. in. The B.H.P. at 1940 r.p.m. is thus raised from 376 to 460, as represented by the point L. If the speed were maintained at 1,940 after the take-off the boost pressure would fall off with the atmospheric pressure, and the power with it, as shown by the line LM. The throttle, meanwhile, would have been fixed in such a position as to give the normal boost of $-\frac{1}{2}$ lb. per sq. in. at 4,400 ft., according to the figure. From M to K the throttle would be gradually opened so as to maintain the normal boost, and the power rises, as before, by reason of the increasing volumetric efficiency and the falling external air temperature.

In practice the r.p.m. would be increased immediately after the take-off to something near the normal, and the power during a climb to the rated altitude would not fall below the point N on the curve of normal boost at normal r.p.m.

It may be observed here that although a variable pitch airscrew is at first sight attractive, and may even be essential on very fast aeroplanes, it would introduce very serious difficulties with highly supercharged engines unless these were provided with some means for disconnecting the supercharger or of reducing its speed ratio to the engine.

The airscrew blades could be set at a fine pitch during the take-off, thus obviating their stalling and allowing the engine to develop its full normal revolutions; but on the combustion side the conditions would be much less satisfactory. The rise of temperature in the supercharger, and the power absorbed by it per lb. of air delivered, both increase nearly in proportion to the square of the speed, and the higher mixture temperature would at the same time promote detonation and reduce the charge weight per cycle at a given boost pressure.

Assuming the same engine and fuel, detonation would make it necessary to use less throttle opening and a lower boost pressure at take-off, and this, combined with the higher air temperature, might very well nullify any advantage from the higher engine speed. It is for this reason that efforts are being made to develop a two-speed supercharger, with a low gear ratio for employment during the take-off and while near the ground, and a high one for pushing up the rated altitude.

It is the necessity of avoiding too high an air temperature in the induction manifolds, rather than design difficulties, that limits the

gear-driven supercharger at the present time to rated altitudes of 10,000–12,000 ft. With a compression ratio at the normal speed of 1·5 to 1, sufficient to maintain atmospheric pressure to 11,000 ft., the rise of temperature in passing through the supercharger is 50° C. and this, added to an external temperature of 15°, produces very severe conditions from the point of view of detonation. The exhaust-driven supercharger, in which the speed can be regulated independently of the engine speed, is from this point of view less objectionable; for the supercharger could be cut out altogether near the ground, or used to give only just the boost which the engine would stand; and at great heights there would be no difficulty about increasing the speed to give really high compression ratios. With the simple impeller with radial vanes coupled direct to the turbine wheel a speed of 30,000 r.p.m. is quite practicable, and an installation of this type has been used in an experimental aeroplane to maintain ground-level pressure up to about 25,000 ft. It is the extreme difficulty of obtaining good mechanical reliability at the high speeds and high temperatures involved in the turbine, and the difficulty also of reducing the danger from fire in the air to something reasonably small, that has prevented the exhaust-driven turbine hitherto from progressing beyond the experimental stage.

The power absorbed by a geared supercharger and the handicap it imposes at take-off are dependent on the rated height for which it is designed, and some comparison of the effects upon the engine of supercharging for different rated heights must now be given. Along the dotted line in fig. 136 are given the h.p. absorbed by four different superchargers at their rated heights of 4,000, 8,000, 12,000, and 16,000 ft. expressed as per 100 B.H.P. of gross engine output. The figures are based on an assumed adiabatic temperature efficiency of 65 per cent. and a mechanical efficiency of 90 per cent. This allowance for gear losses is based on tests[53] of superchargers for engines of about 500 B.H.P. It was stated in Chapter X that a mass-flow of air of 1 lb. per sec. would suffice for a well-tuned engine of about 540 B.H.P.

In order to provide the correct compression ratio for each different rated height the gearing between the crankshaft and impeller would be such as to give the necessary tip-speed at the normal r.p.m. of the engine (see the calculation on p. 269). The power absorbed varies according to the gear ratio, and from the rated height downwards would be substantially a constant fraction of the B.H.P. This follows from the fact that the work per lb. of air would be constant at constant speed, and any change in the mass-flow would be nearly in proportion to the B.H.P. Any change of speed from the normal

r.p.m. would alter the work per lb. of air in proportion to N^2 and the power absorbed in proportion to N^3, since the mass-flow of air is proportional to N.

The rise of the air temperature in passing through each supercharger, and the induction manifold temperatures at ground-level and at the rated heights, are given by the other three lines in fig. 136. It will be seen that at its normal speed the 12,000-ft. supercharger

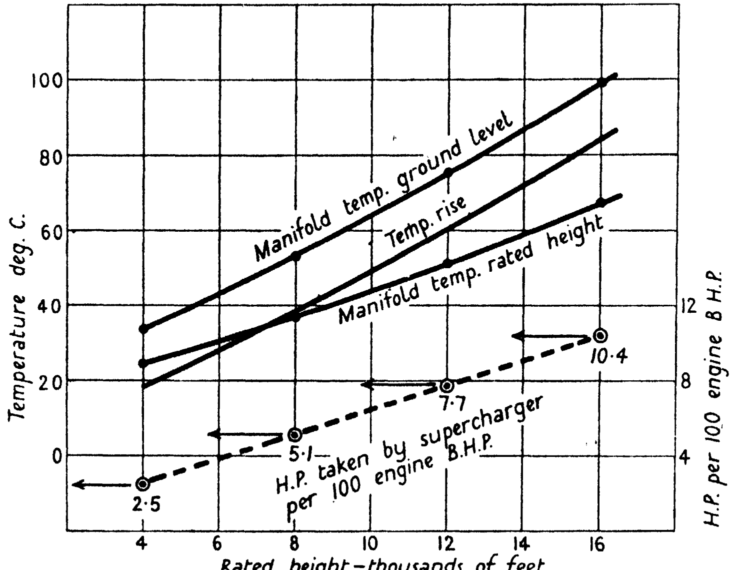

FIG. 136. H.P. absorbed, and temperature of air delivered by superchargers which maintain ground-level pressure to 4, 8, 12, and 16 thousand feet.

would produce an induction-pipe temperature at the ground of 75° C., and would absorb 7·7 per cent. of the engine B.H.P. Under take-off conditions the speed would be lower, and allowance must be made for this in estimating the power absorbed. The rise of temperature would also be lower, and would correspond to the compression ratio at the take-off speed, but even so this represents about the limit for which the single-stage supercharger without intercoolers can be designed.

ART. 78. *Ground-level tests of supercharged engines.*

When an engine is designed with a supercharger which maintains a constant pressure in the manifold up to the rated altitude, so that its highest power at the normal speed is obtained at that height, then an actual measurement of the maximum power could only be made

at ground-level if means were available not only for maintaining in the exhaust manifold the atmospheric pressure at the rated altitude, but also for maintaining the inlet air at the correct pressure and temperature. The correct pressure, of course, presents no difficulty, for that can be got by throttling; but the cooling of the air necessary for a 500 h.p. engine (about 55 lb. per min., or 725 cu. ft. at N.T.P.) is a formidable problem involving large and expensive plant, apart altogether from the large exhauster pumps necessary for maintaining the low pressure on the exhaust side. A matter of special difficulty in connexion with the cold air supply is that of getting rid of the snow which forms when air of normal humidity and temperature is cooled to well below zero.

In these circumstances it is clearly necessary to test supercharged engines in general at ground-level without these special arrangements, and to have reliable corrections to apply in order to calculate the powers which will be available at the rated altitude.

If an engine has a declared rated altitude of, say, 10,000 ft., then the straightforward test at ground-level would be one at the normal speed in which the pressure at the inlet to the supercharger was maintained at 10·1 lb. per sq. in., which is the pressure at 10,000 ft. in the standard atmosphere. Having measured the B.H.P. in such a test, the inlet air temperature being, say, t° C., the following corrections would then have to be applied to obtain the true B.H.P. available at the rated altitude.

(*a*) To the supercharger.

At the normal speed the supercharger would give a certain observed compression ratio, but this would be with inlet air at temperature t° C., and the effect of a lowering of the temperature to that in the standard atmosphere at 10,000 ft., $-4{\cdot}8^\circ$ C., would be to increase the pressure ratio. Reference to Chapter X shows that the compression ratio of a centrifugal supercharger is related to the air temperature according to the equation

$$R^{(\gamma-1)/\gamma} = 1+\frac{\eta W}{K_p T_1}.$$

For computation purposes it has been found that the effect of the inlet air temperature can be represented with sufficient accuracy, over the range of temperature required, from about $+15^\circ$ to -10°C., by the empirical formula

$$r_z = r_0[1+0{\cdot}00063r_0^2(t_0-t_z)], \tag{83}$$

in which r_0 is the observed pressure ratio at ground-level and t_0 is the air temperature there, while t_z and r_z refer to the rated altitude.

The 'rated boost pressure' given by the supercharger at the rated altitude will be somewhat greater than the observed boost pressure at ground-level, in the ratio r_z/r_0, and the engine power must be corrected accordingly as follows below.

(*b*) Corrections to the engine power.

At the rated altitude the engine would give more power than that observed in the ground-level test for three reasons:

(1) On account of correction (*a*) above, the pressure in the induction pipe would be greater than the boost pressure at ground-level in the ratio r_z/r_0, and the B.H.P. may be assumed to be increased in proportion to the small change of pressure involved.

(2) The temperature of the air delivered would be lower by (t_0-t_z), where t_0 is the observed ground-level temperature and t_z is that at the rated altitude in the standard atmosphere. We have seen that the I.H.P. is inversely proportional to $\sqrt{\theta}$, and it is sufficiently accurate for practical purposes when making this correction to assume the B.H.P. to increase in the same ratio.

(3) The back pressure in the exhaust system is lower by the amount of the fall in atmospheric pressure up to the rated altitude. The effect of a change of the exhaust back pressure upon the B.H.P. of an engine will depend a good deal upon the speed and the valve timing; and the magnitude of the effect will be different according to whether the back pressure is above or below that in the inlet manifold. Bench tests of a typical 12-cylinder engine[67] at 2,000 r.p.m. have shown that the relationship was linear and that a rise in the exhaust back pressure of 1 lb. per sq. in. had substantially the same numerical effect on the I.M.E.P. at all heights, varying only from 2·7 lb. per sq. in. at ground-level to 3·4 at 15,000 ft. ($p = 0{\cdot}564$). This is equivalent to a fall of 2 per cent. in the ground-level B.H.P. for each 1 lb. per sq. in. rise in the exhaust manifold. When the exhaust pressure is below the inlet pressure the increase of the B.H.P. is a little less rapid than this. It amounts very closely to 1½ per cent. of the ground-level B.H.P. for each 1 lb. per sq. in. drop in the exhaust manifold pressure, or 1 per cent. drop for each 35 mm. fall in pressure below 760 mm. A more accurate empirical formula for the correction factor, which takes into account the mechanical efficiency of the engine at ground-level (without the supercharger), is as follows:

$$\text{correction factor} = \frac{(1+0{\cdot}005\delta p_z)(1+0{\cdot}0125\eta_m\,\delta p_z)}{(1+0{\cdot}005\delta p_0)(1+0{\cdot}0125\eta_m\,\delta p_0)}.$$

In this formula δp_0 is the algebraic difference of pressure between

the inlet and the exhaust manifolds during the ground-level tests and δp_z is the same quantity at the rated altitude. η_m is the mechanical efficiency. δp_0 and δp_z are measured in lb. per sq. in.

As a numerical example of the size of these three corrections to be applied to the observed power at ground-level, there follow some test figures which would be typical of the engine referred to in the last article. The stated and observed data may be given as

Rated altitude . . .	. 10,000 ft.
Barometer . . .	. 29·9 in.
Depression at air intake	. 9·3 in. (corresponds to 9,900 ft.
∴ pressure ,, ,,	. 20·6 in. = 10·1 lb. per sq. in.
,, in induction pipe	. 28·9 in. = 14·2 ,, ,,
Air temp. at intake .	. 15° C.
Observed B.H.P. .	. 430.

Correction (*a*) to supercharger:

$$t_z = -4{\cdot}6^\circ\text{ C. and } (t_0 - t_z) = 19{\cdot}6,$$

$$r_0 = \frac{14{\cdot}2}{10{\cdot}1} = 1{\cdot}400,$$

whence $r_z = 1{\cdot}435$,

and the boost pressure at 9,900 ft. would be 14·55 instead of 14·2 lb. per sq. in. as observed.

Corrections (*b*) to observed B.H.P. of engine:

(i) correction factor $r_z/r_0 = 1{\cdot}024$;

(ii) ,, ,, $\sqrt{\dfrac{273+15}{273-4{\cdot}6}} = 1{\cdot}036$;

(iii) ,, ,, from formula $= 1{\cdot}076$
(1·07 by approx. rule).

Total correction factor $1{\cdot}024 \times 1{\cdot}036 \times 1{\cdot}076 = 1{\cdot}140$.

Corrected B.H.P. $= 430 \times 1{\cdot}140 = 490$ at 9,900 ft.

XIV

FUEL ECONOMY

ART. 79. *Cylinder temperature as the controlling factor in fuel economy.*

The point has been emphasized repeatedly in the present volume that when the power output per cylinder of an engine is raised to the highest limit in order to reduce the weight per h.p., then the problem of dealing with the waste heat is always the critical one. Breakdown will normally take the form of overheating either of the piston or valves, or of a lubrication failure which can be traced to the same ultimate cause. One may go so far as to say that if an engine is *not* within a narrow margin of overheating at some point or other it is not working at its maximum capacity; or, expressed otherwise, that a change of condition which reduces the cylinder temperatures at critical points under any given conditions of running, will raise the limit of power output of which the cylinder is capable.

It is well known from accurate experiments on single-cylinder engines that the maximum power output can be obtained from a cylinder when the fuel-air mixture is about 20 per cent. rich. This follows from the chemistry of combustion of the mixture, and is independent of the cylinder conditions. So also experiments on test engines have shown that the maximum thermal efficiency and minimum fuel consumption per I.H.P. hour is obtained when the mixture is 15–18 per cent. weak. This again is the result of the conditions of combustion and is found to be the same in any normal cylinder.

When considering fuel economy in an aero-engine, however, and especially in an air-cooled engine, one's point of view has to be entirely altered from that of regarding the necessary adjustment to get a high economy as merely one of obtaining the right mixture strength. The reason is that every alteration of mixture strength causes a change of cylinder temperature. A weakening of the mixture strength below that which gives maximum power leads at first to substantially higher temperatures of the cylinder and exhaust valves, and in consequence all adjustments with a view to economy in fuel have to be made in relation to this question of cylinder and valve temperatures and of possible damage through overheating.

The relationship between cylinder temperatures and mixture strengths as found in certain aero-engine cylinders will be given in art. 81, and in the light of that data the limits of fuel economy

in practice will be discussed. As a background to that discussion it is useful to keep in mind some typical figures for the minimum fuel consumption per h.p. as obtained in a test engine. These are given in fig. 137 for compression ratios between 4 : 1 and 7 : 1, using an average petrol of specific gravity 0·76, at 15° C., and lower calorific value 10,500 C.H.U. per lb.

The fuel consumption per I.H.P. hour is the only basis upon which engines can be scientifically compared, but, on the other hand,

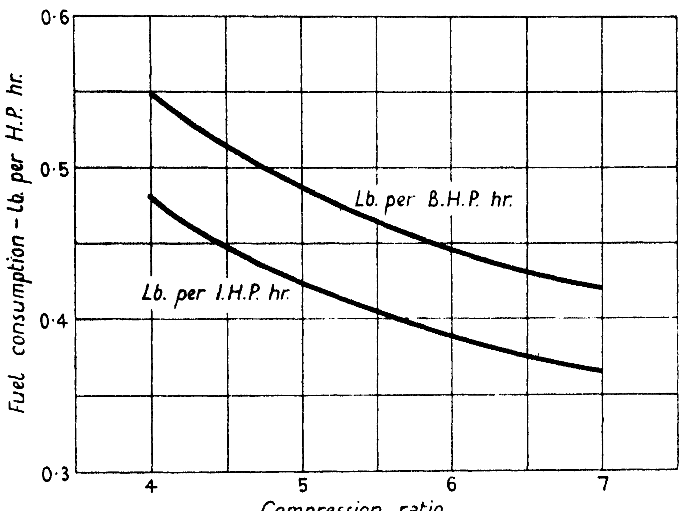

FIG. 137. Rates of minimum fuel consumption per I.H.P. hour at different compression ratios as found in single-cylinder tests; and the corresponding figures per B.H.P. hour assuming a mechanical efficency 87 per cent.

the consumption per B.H.P. hour is the important figure in practice, and the one more usually available. A curve of consumptions per B.H.P. hour has therefore been added to fig. 137. This does not represent the consumptions actually observed on the test engine but are those calculated from the figures on the I.H.P. basis with an assumed mechanical efficiency of 87 per cent. All aero-engines under cruising conditions at moderate heights (when fuel economy is important) would show a mechanical efficiency within 1 or 2 per cent. of this, rising to about 90 per cent. at full throttle and normal full speed. The upper of the two curves in fig. 137 therefore gives a fair datum which the multi-cylinder engine should approach under cruising conditions if the equality of fuel distribution between the cylinders were perfect and if all were working with a fuel-air mixture 15 per cent. weak. It is useful to have these optimum figures for each

whole number compression ratio available for reference, and they have therefore been collected together in table 52.

TABLE 52

Fuel consumptions per B.H.P. hour which should be obtainable on a single-cylinder engine of mechanical efficiency 87 *per cent. when employing a fuel of calorific value* 10,500 *C.H.U. per lb.*

Compression ratio	*Fuel consumption lb. per B.H.P. hour*
4	0·55
5	0·485
6	0·445
7	0·42

ART. 80. *The relation between fuel consumption and power output.*

In fig. 63 (i) there were given examples of 'consumption loops' obtained on a 4-cylinder engine at three different throttle positions, all at the constant speed of 1,000 r.p.m. Those curves showed how the fuel consumption per B.H.P. hour varied in relation to the power at constant speed and constant throttle position while the fuel-air mixture was weakened, step by step. When considering the possibilities of fuel economy in the air it is important to have before us the results of corresponding tests showing the rate of fuel consumption in relation to power output, but the conditions of the tests must be chosen by reference to the conditions in flight, and a series of consumption loops all at the same speed, as in fig. 63 (i), would not give the information required. Moreover, in addition to the relation between the power and fuel consumption at different fuel-air ratios it will be necessary, for the reasons given in the last article, to know the simultaneous variations of the cylinder temperatures.

In art. 71 it was explained that as an engine is throttled down from the condition of full speed in level flight to a normal cruising speed there is a definite relation, given by the curve in fig. 123 and nearly the same for all aircraft, between the B.M.E.P. and the r.p.m. For a study of fuel economy in flight it is required to know the lowest practicable fuel consumption per B.H.P. hour at a series of forward speeds, and this data must be obtained from several different series of tests, each series at a constant speed and at the constant B.M.E.P. which goes with that speed in level flight. During each series the fuel-air ratio must be weakened progressively and the throttle opened slightly to keep up the B.M.E.P. with the weaker mixtures.

In some recently published tests,[60] of a 9-cylinder radial engine of which the maximum permissible speed was 2,200 r.p.m., consumption loops were taken at 2,010, 1,900, 1,700, and 1,400 r.p.m. similar to those in fig. 63 (i) in that throughout each loop the throttle position was kept constant and the B.M.E.P. was allowed to fall off as the mixture strength was weakened. The loops are reproduced in fig. 138, and the beneficial effect of increasing the ignition advance

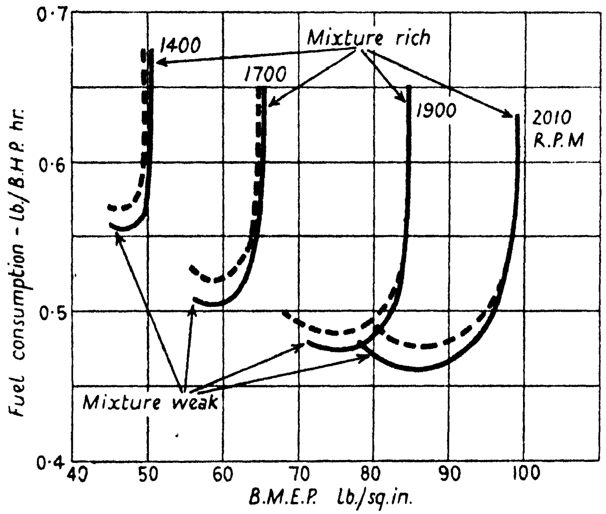

FIG. 138. Fuel consumption loops for 9-cylinder radial aero-engine, air-cooled. Compression ratio 5·8. R.P.M. and B.M.E.P. at minimum of each loop related as in level flight by adjustment of the throttle before the weakening of the fuel-air mixture.

as the mixture was weakened should be noticed. The dotted curves were obtained with a fixed ignition advance of 38 deg., while the full lines show the result of increasing the advance so as to obtain the maximum power at each reading, which occurred with ignition timings varying from 38 deg. up to a maximum of 50 deg. with the weakest mixtures at each throttle setting.

The B.M.E.P.s corresponding to engine speeds of 2,010, 1,900, 1,700, and 1,400 r.p.m. in level flight were 84·5, 75, 61, and 45 lb. per sq. in., and curves like those of fig. 138 do not therefore give the minimum consumption for level flight unless the lowest points of the loops should happen to coincide with these values of the B.M.E.P. In the trials referred to, each consumption loop was started at a B.M.E.P. sufficiently above that corresponding to level flight to produce approximately the correct value at its lowest point. Thus at 1,700 r.p.m. the loop was started at 65 lb. per sq. in., at which value

the consumption might be anything from about 0·56 to 0·65 lb. per B.H.P. hour, and by the time the minimum consumption of 0·505 was reached the B.M.E.P. had fallen to 59, as compared with the required value of 61 lb. per sq. in. At 1,900 r.p.m. the coincidence was exact, at 75 lb. per sq. in.

It must not be assumed that it would be either safe or desirable to cruise with the air-fuel ratio giving the minimum consumption. Before that question could be settled the effect of the weakening of the mixture upon the cylinder and valve temperatures would need to be examined, as well as upon the sparking plugs and other essential elements in the reliability of the engine. As mentioned in the last article, the normal reliability of an engine may be completely destroyed if abnormal temperatures are permitted.

For the study of the effect of the air-fuel ratio upon cylinder temperatures and for showing the relation between the fuel consumption and the permissible conditions in practice, the tests must be of the type mentioned earlier, in which a series of power observations with varying air-fuel ratio is made at various values of the r.p.m., and throughout the series the B.M.E.P. is maintained constant. With the engine running at the particular speed required, the throttle must be closed until the correct B.M.E.P. for level flight is obtained with a rich air-fuel mixture. The mixture control is then operated so as to give a weaker mixture, and as soon as the engine shows any fall of B.M.E.P. due to this weakening, the throttle is slightly opened until the original B.M.E.P. and r.p.m. are restored. The mixture is then weakened further, step by step, and at each step the engine is given a little more throttle sufficient to keep a constant B.M.E.P. throughout the series.

It will no longer be possible to plot the fuel consumption per B.H.P. hour on a basis of B.M.E.P., as in fig. 138, and a convenient form of diagram is now to plot it against the mixture strength, as in fig. 139, or against the ratio of air to fuel in the mixture. In this way a series of fuel-consumption curves at constant power can be obtained similar to that shown in fig. 139, each one corresponding to a certain speed and its associated B.M.E.P. for level flight. The lowest point of each curve will then show the minimum fuel consumption of which the engine is capable at that speed, but the question of how far towards the weak mixture end of the curve it will be safe to go, for normal cruising conditions, can only be settled in relation to the reliability of the engine. In the trial during which the results of fig. 139 were obtained the limit of steady running of the engine was reached when the mixture was only 5 per cent. weak, so that, apart from temperature effects in the engine, the vibration in the

aeroplane caused by rough running would probably preclude the use of any mixture weaker than that.

A valuable indication of the severity of the conditions to which an engine is being subjected can be obtained from the temperature readings of thermocouples placed in the cylinder heads, and in the next article are given the results of such measurements made throughout a series of tests of an air-cooled engine at constant speed and power, when the air-fuel ratio, and hence the fuel economy of the engine, was varied.

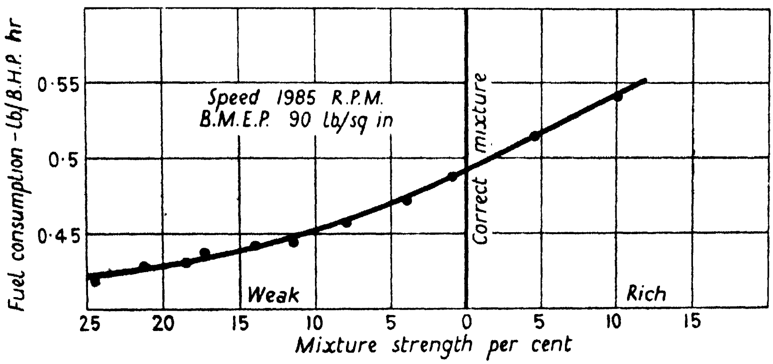

FIG. 139. Variation of the fuel consumption per B.H.P. hour of a 9-cylinder air-cooled aero-engine when the air-fuel ratio was varied under conditions of constant power output. Compression ratio 5·8. Speed 1,985 r.p.m. B.M.E.P. 90 lb. per sq. in.

ART. 81. *Mixture strength and cylinder temperatures.*

Temperatures at the surface of the combustion space of an air-cooled cylinder at full load will vary from a maximum of about 750° C. on the exhaust valve down to 350° at the hotter and 200° at the cooler parts of the cylinder head itself. Any figure quoted for the 'cylinder temperature' under certain conditions is therefore of purely comparative value, and does not pretend to be an average temperature representative of the whole head. In what follows, the temperature readings are those which are obtained from a thermocouple deeply embedded in a bronze plug screwed into a hole in the aluminium alloy head of an air-cooled cylinder at a point between the two exhaust valves. The temperatures are therefore approaching those of the hottest parts of the head itself, but below those of the exhaust valve and piston centre.

The temperatures observed on an air-cooled engine will depend not only on the air-fuel ratio, the speed, and the throttle opening, but at any given value of these variables they will, of course, depend entirely on the velocity of the cooling air-stream maintained over the

cylinders. Since the object of the tests about to be described is to determine what air-fuel ratio can safely be adopted in flight, the conditions in regard to the cooling air during the tests must reproduce those of flight so far as possible. We have already seen that, as an engine is throttled down from full speed in level flight, each value of the r.p.m. will be associated with one definite value of the B.M.E.P., as it will be also with a definite speed of the aeroplane, and hence of the cooling air. It follows, therefore, that while making a test of the effect of the air-fuel ratio on cylinder temperatures, the cooling air over the engine must be adjusted for each value of the power output in the same way as it would vary in flight.

It has been found that the B.H.P. and forward speed in level flight are closely related by the equation

$$\text{B.H.P.} \propto V^{2\cdot 8},$$

and in the tests described below the cooling air velocity was adjusted accordingly. Throughout a series of tests at a given r.p.m. the cooling air was constant, and the only variable was the air-fuel ratio.

The temperature variations observed on a 9-cylinder air-cooled engine very similar to the one used for the tests of fig. 138 are shown in fig. 140. The five curves refer to the five values of the power and the cooling air velocities given in table 53. The curves are only to

TABLE 53

Corresponding values of the power, speed, and cooling air velocity for each series of trials illustrated in figs. 140 *and* 141.

Max. permissible r.p.m. 2,100.

Engine r.p.m.	*B.M.E.P. for level flight*	*B.H.P.*	*Cooling air speed m.p.h.*
1,985	90	395	126
1,865	80	330	119
1,730	70	268	111
1,575	60	209	103
1,365	50	151	93

be taken as generally indicative of what is to be expected when weak fuel-air mixtures are employed, for such thermocouple readings vary from one cylinder to another, and are much influenced by the ignition timing, and by cowling round the cylinders. In the present series of tests there was no cowling, and the ignition timing was adjusted, for each mixture strength, to the optimum, i.e. to the maximum power value.

It will be observed that, although the maximum gas temperatures

would have been reached at a mixture about 15 per cent. rich, the observed temperatures, and probably also those of the pistons and exhaust valves, reached maximum values at much weaker mixtures, owing to the influence of a slower rate of burning and of high gas temperatures during the exhaust stroke.

The relation between the cylinder temperatures and fuel economy for the same series of tests is shown more explicitly in fig. 141 where

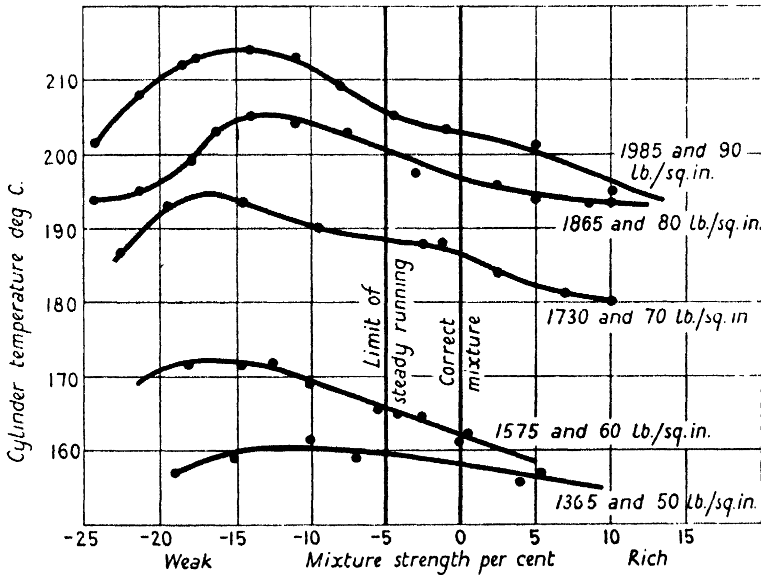

FIG. 140. Observed variations of cylinder temperature with mixture strength under conditions of constant speed and power output. 9-cylinder air-cooled aero-engine. Cooling air velocity varied in proportion (B.H.P.)$^{1/3}$. Compression ratio 5·8.

the observed temperatures are plotted against the fuel consumption in lb. per B.H.P. hour. The approximate rates of fuel consumption at which unsteady running became noticeable at each speed are marked on the curves. As a tentative conclusion these points might be taken as the limits to which it would be safe to go in the direction of weakening the mixture for the sake of fuel economy; but it must be understood that the ability of the engine to sustain, for example, a B.M.E.P. of 90 with a fuel consumption of 0·47 lb. per B.H.P. hour with full reliability would have to be proved out by a prolonged endurance test of 100 hours or more.

Subject to this proviso, the data given in fig. 141 provides a complete picture of the fuel economy actually achievable in flight near the ground, and the ideal carburettor would maintain that mixture

strength at each position of the throttle which gives the maximum economy without overstepping the limit of steady running.

Art. 82. *Unsteady running on weak mixtures.*

The limits of economy attainable in flight for the 9-cylinder radial engine, as defined by the limits of steady running, are seen from fig. 141 to range from 0·47 to 0·49, according to the degree of throttling, except for the lowest B.M.E.P. of 50 lb. per sq. in.

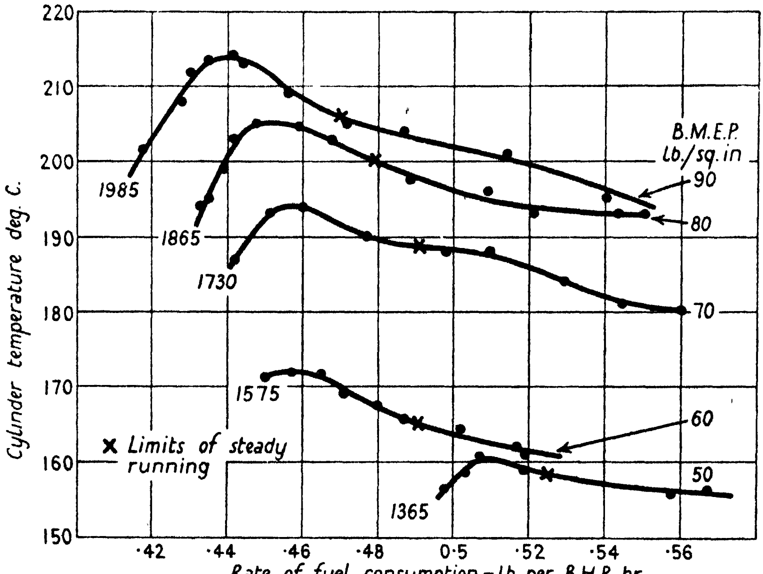

Fig. 141. Variation of cylinder temperature with fuel consumption per B.H.P. hour under conditions of constant speed and power output.

Temperature conditions at this low power would be unimportant, and for practical purposes consideration may be limited to cruising conditions at a B.M.E.P. of 70 and over. In these circumstances it appears that a fuel consumption of the order of 0·48 lb. should be possible, as compared with about 0·56 at full throttle. An account[60] has been given of an endurance test of 100 hours at 2,000 r.p.m. with an engine of the same type at a B.M.E.P. of 85 lb. sq. in. (396 B.H.P.) and a mean fuel consumption of 0·481 lb. No serious failure occurred, although the running at a mixture weak enough to give this low rate of consumption combined with an output of 396 h.p. was found to be very severe on the sparking plugs, and to a lesser degree on the exhaust valves and valve guides.

The unsteadiness of running experienced when the mixture was

weakened more than about 5 per cent. below the correct air-fuel ratio arises from the rapid change of mean pressure with mixture strength on the weak side. The problem has been thoroughly explored in two further sets of trials recently published,[47] on an air-cooled and a water-cooled engine, which deserve the most careful study. They include comparative performance and fuel-consumption tests when using petrol and also vaporized butane as fuel, with a view to studying the importance of inequalities in the distribution of

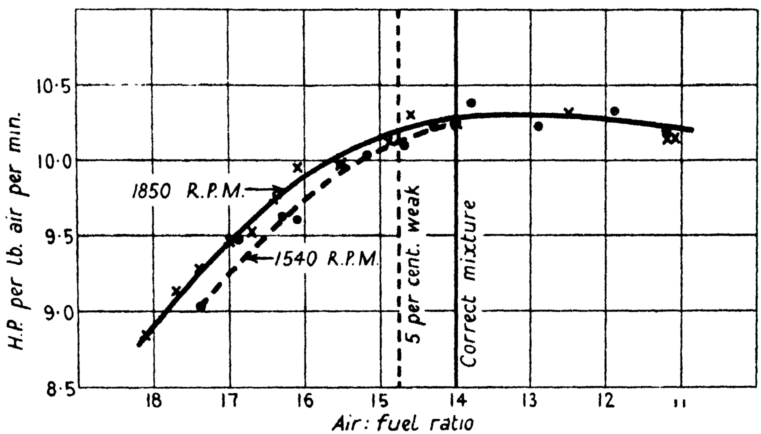

FIG. 142. Variation of I.H.P. per lb. air per min. with mixture ratio at 1,850 and 1,540 r.p.m. in an air-cooled 9-cylinder engine of compression ratio 5·8.

liquid fuel to the cylinders, both in regard to the average fuel consumption and to the smooth running of the engine. This would be expected to be more serious in the water-cooled engine with its less symmetrical induction system, and it was on this engine that the difference between the two fuels was most marked, for the inequalities of distribution were reduced, if not eliminated with the vaporized butane. It was found that the limit of weakness to which steady running was maintained was always greater with the gaseous fuel, but not very much so, and the conclusion drawn as regards the relative fuel consumption was that, with some variation according to the pressure and temperature of the air supply, the inequality of fuel distribution was responsible for an increase in the consumption of the liquid over the gaseous fuel, under comparable conditions, of not more than 5 per cent.

Although the average fuel consumption per B.H.P. for all the cylinders might be increased only 5 per cent., this would be quite enough to allow of substantial variations between the M.E.P. and the power delivered by individual cylinders. That this is so may be

understood by reference to the two curves in fig. 142. These show the variation with mixture strength of the I.H.P. per lb. of air per min. for two tests on the air-cooled engine. The tests took the form of consumption loops, similar to those of fig. 138, at 1,850 and 1,540 r.p.m.

The ordinates are proportional to the mean gas pressure in all the cylinders, for at constant speed and constant throttle position the rate of air consumption would be very nearly constant throughout each test. There was, in fact, an increase of a little over 1 per cent. between the weak and the rich ends of the curve. It is clear from the figure that so long as the mixture was rich, small variations of the air-fuel ratio would not affect the M.E.P. of individual cylinders, but that as soon as the average ratio was greater than 14 : 1 quite small inequalities of fuel distribution would produce appreciable variations of M.E.P. in different cylinders, and these account for the roughness of running which became evident at values of the air-fuel ratio greater than 14·7. For suppose, first, that the distribution were perfect and every cylinder receiving the 'correct' mixture; and then that for some reason one cylinder received a mixture 20 per cent. weak. If we suppose the total fuel supply to remain the same, so that the other cylinders were slightly enriched, this would not affect their M.E.P.s, as is clear from the figure, but the power of the one weak cylinder would be down just 10 per cent. This in itself would be sufficient to account for the onset of rough running, and yet the average fuel consumption per B.H.P. for the whole engine would have increased by only 1·1 per cent.

The trials upon the water-cooled engine in the same report include fuel consumption measurements under altitude conditions up to 23,000 ft., and will be further considered in the next article.

Before passing on to the question of altitude and fuel economy it will be useful to summarize the results of the complete engine trials so far considered, and to compare them with the single-cylinder results given in fig. 137. All the trials reviewed in the last two articles have been upon an engine of compression ratio 5·8, at which value the curves of fig. 137 show a consumption of 0·395 lb. per I.H.P. hour and a rough figure of 0·45 lb. per B.H.P. hour with the assumed mechanical efficiency of 87 per cent. As compared with these figures, the consumption loop in fig. 138 for 2,010 r.p.m. shows a minimum consumption at nearly full throttle of 0·46 lb. per B.H.P., and in the more recent report[47] the test upon the same type of engine at 1,940 r.p.m. gave 0·44.

The differences between the minima in these two reports represent about the order of agreement to be expected between different

sets of trials and are due to slight differences of tune in the engines. It must be remembered that, apart from the technical difficulty of carrying through an accurate set of trials such as those referred to, the points of minimum fuel consumption always fall within the region of unsteady running, and this accentuates the difficulty of repeating the experimental conditions very closely. At what air-fuel ratio the unsteadiness may become serious depends very largely, for example, on the condition of the sparking plugs. It was found that when working on the weak mixtures a plug, which would normally last 100 hours or more, might retain its full efficiency for no more than 3 hours. After this short time, although it would function satisfactorily with normal mixture strengths, misfiring was liable to occur with the very lean ones.

ART. 83. *Altitude and fuel consumption.*

The effect of varying the mixture strength upon the I.H.P. derived from every lb. of air consumed by an engine was illustrated in fig. 142 of the last article. The relation between the thermal efficiency of an engine and its power per unit of air consumed has been dealt with in art. 40 (i), and all that we are now concerned with is the fact that at a given air-fuel ratio, or, more precisely, at a given heat value per cu. ft. of combustible mixture, the I.H.P. per lb. of air will depend only on the thermal efficiency of the engine; and that over the range of engine speeds which are important in flight this may be expected to remain very nearly constant at all heights, provided always that the fuel is properly vaporized and the ignition timing adjusted for maximum power.

The test results quoted in the last article lead to the conclusion that an average air-fuel ratio equal to that in the chemically correct mixture is very close to the most economical one which smooth running will allow, and for practical purposes, therefore, this air-fuel ratio is a good one to aim at as representative of economical cruising conditions, when discussing the effect of altitude upon fuel consumption.

At full throttle near the ground, or up to the rated altitude of a supercharged engine, rich mixtures will be used for the sake of the extra power obtained and of the cooler cylinder conditions, and under these conditions good economy is not to be looked for; but for M.E.P.s of less than three-quarters of the maximum full-throttle value it should be possible to operate safely at the correct air-fuel ratio, and under these conditions the I.H.P. of an engine may be expected to bear a constant ratio to the air consumed.

The two series of trials referred to in arts. 80–82 show that the

I.H.P. per lb. of air per min. at the correct mixture did in fact remain very nearly constant under all conditions. For the tests represented in fig. 138 the data are scarcely sufficient to get an accurate figure for each test, but the mean estimated value for the four consumption loops in fig. 138 is 10·1, and from another series of tests on the same engine at five different throttle openings, all at 2,000 r.p.m., the value was also 10·1 with an extreme variation of $\pm\frac{1}{2}$ per cent.

For the two engines in the later report[47] the complete results are summarized in table 54. The mean value of the I.H.P. per lb. of air

TABLE 54

Observed values of the I.H.P./lb. of air/min. at the correct fuel-air ratio with petrol and a gaseous fuel.

Air-cooled engine. C.R. = 5·8			*Water-cooled engine. C.R. = 6*						
	Fuel			*Ground-level*		*15,000 feet*		*23,000 feet*	
Speed	*Petrol*	*Butane*	*Speed*	*Petrol*	*Butane*	*Petrol*	*Butane*	*Petrol*	*Butane*
1,940	10·2	10·2	2,028	10·5	10·5	10·2	10·0	10·5	10·0
1,850	10·25	10·25	1,920	10·6	10·7	10·1	9·9	10·1	10·1
1,780	10·4	10·4	1,804	10·6	11·0	10·3	10·3	9·6	9·5
1,540	10·3	10·5	1,576	10·8	11·2	10·2	10·0	..	10·0
1,410	10·15	10·3		*Ground-level*		*15,000 feet*		*23,000 feet*	
Mean for a.-c. engine, ground-level, 10·3			Mean	10·7		10·1		10·0	

for the air-cooled engine over the whole range of speeds and throttle settings and on the two different fuels is 10·3, with an extreme variation of +2 and −1·5 per cent. The mean value from all the ground-level tests on the water-cooled engine, with its somewhat higher compression ratio, is 10·7, and the average value falls to 10·1 and 10·0 for 15,000 ft. and 23,000 ft. respectively. The air-cooled engine was not tested under altitude conditions.

Considering the wide variety of the conditions covered and the difficulty of achieving perfect consistency in full-scale trials of this magnitude, the uniformity of the results in table 54 is very satisfactory; and one is justified in concluding that variations from a mean value of the I.H.P./lb. air/min. in any one engine at the correct mixture strength are not due to anything fundamental, but simply to variability in one or more of the experimental conditions, such as the state of the sparking plugs or uniformity of the fuel-air ratio. In the trials of the water-cooled engine, the whole series at 9,600 ft. was vitiated by irregular changes in the air and petrol flow, caused by small deposits of ice or snow in the carburettor and induction system.

The fact that the I.H.P./lb. air/min. in the water-cooled engine dropped from 10·7 at ground-level to 10·1 at 15,000 ft. may be interpreted as due to more erratic distribution of the fuel between the cylinders, and even non-uniformity of the fuel-air ratio within a cylinder, when the air temperature was low and evaporation sluggish; for the temperature of the air supply in these trials was lowered to correspond to conditions in the standard atmosphere.

Having established the constancy of the I.H.P./lb. air/min. under all conditions of height and throttle opening, provided no special conditions are present to upset the proper functioning of the engine, we then have a firm basis on which to make an estimate of the fuel consumption per B.H.P. hour under any condition in flight, for the step between the one and the other is reduced to the question of how the L.H.P. varies with height and throttle.

It is necessary to fix certain flight conditions under which to compare, for example, the fuel consumption at different heights, and these would most naturally be specified by making the comparison at the same A.S.I. reading in level flight, for this means that the airscrew efficiency and the engine torque and B.M.E.P. would also be the same.

Equation (73) on p. 337 was shown to give values of the L.M.E.P. consistent with the tests of the 9-cylinder normally aspirated engines so far considered. Another engine of a closely related type was used in the range of flight trials referred to more fully in the next article, and, although motoring loss tests are not available on that particular engine to check the equation, there is little doubt that it may safely be used for estimating the L.M.E.P.

It is proposed, therefore, in the present article to undertake as a numerical example the calculation of the fuel consumption per B.H.P. hour for the air-cooled radial engine of the flight trials[66] at 5,000 and 15,000 ft., and in the following article to compare the result with the figures obtained during the trials in the air. The estimated figures for the rate of consumption at ground-level have also been included for the sake of completeness.

A convenient A.S.I. reading at which to make the comparison is that which gave maximum range for the aeroplane, and is the same for all heights, namely 75 m.p.h. At this speed the engine r.p.m. as observed in flight were 1,375 at 5,000 ft. and 1,600 at 15,000.

We start from the facts that with a non-detonating fuel the engine, at ground-level, gave a B.M.E.P. of 127 lb. sq. in. at 1,780 r.p.m. and that in flight the airscrew was such as to allow a maximum rotation rate of 1,870 r.p.m. at 5,000 ft. From the relationship between torque in level flight and r.p.m., illustrated in fig. 123,

it follows that when, at 5,000 ft., $V_i = 75$ and $N = 1{,}375$, the B.M.E.P. was 60 lb. sq. in., and was the same at 15,000 ft. at 1,600 r.p.m. The r.p.m. near the ground at the same A.S.I. reading would have been in proportion to $\sqrt{\sigma}$ and equal 1,280.

The calculation of the fuel consumption per B.H.P. hour is set out, step by step, in table 55 and calls for no explanation except to

TABLE 55

Fuel consumption per B.H.P. hour for a 9-cylinder air-cooled engine, normally aspirated, calculated on the assumption of the correct mixture and an air consumption corresponding to 10·3 I.H.P./lb. air/min. under all conditions.

Quantity	Remarks	Ground-level	Height 5,000 ft.	Height 15,000 ft.
Pressure ratio	in stand. atm.	1	0·832	0·565
Pressure	in stand. atm.	14·7	12·24 lb. sq. in.	8·30 lb. sq. in.
Density ratio	in stand. atm.	1	0·862	0·629
B.M.E.P.	full throttle	127	106	72·6
r.p.m.	full throttle	..	1,870	1,760
Manifold pressure	full throttle	14·2	11·85 lb. sq. in.	8·10 lb. sq. in.
B.M.E.P.	at $V_i = 75$	60	60 „ „	60 „ „
r.p.m.	at $V_i = 75$	1,280	1,375	1,600
Manifold pressure	at $V_i = 75$	6·7	6·4 lb. sq. in.	6·0 lb. sq. in.
L.M.E.P.	from equation (73)	10·6	10·1 „ „	9·4 „ „
I.M.E.P.		70·6	70·1 „ „	69·4 „ „
I.H.P.	cyl. size $5\frac{3}{4} \times 7\frac{1}{2}$	200	214	246
Air lb./min.	at 10·3 I.H.P./lb./min.	19·4	20·8 lb.	23·9 lb.
Fuel lb./min.	air/fuel = 14	1·39	1·49 „	1·71 „
Fuel consumption lb./B.H.P. hour		0·491	0·489	0·482

point out that the full-throttle B.M.E.P.s at 5,000 and 15,000 ft. are calculated by equation (78) (p. 343) from the ground-level value. Manifold pressures are taken as proportional to B.M.E.P.s, with allowance, under throttled conditions, for the falling exhaust pressure above ground-level. The L.M.E.P. due to mechanical friction and pumping loss is calculated from equation (73). The full speed N_0 of the engine has been taken as 2,200 r.p.m.

It will be observed that, in spite of the increase of mechanical friction which accompanies the increase of r.p.m. from 1,280 to 1,600 between ground-level and 15,000 ft., the total L.M.E.P. is 1·2 lb. sq. in. less at 15,000 ft. by reason of the reduction of the pumping loss. This diminishes on account both of the lower atmospheric pressure and of the more fully open throttle at 15,000 ft.

when $V_i = 75$ m.p.h. The diminution of the L.M.E.P. should produce a reduction of the fuel per B.H.P. hour amounting to 2 per cent., as between ground-level and 15,000 ft., and a proportionate increase of the range in still air, provided always that the same mixture ratio is used and that an ignition advance can be given which is sufficient to produce maximum power at each height and throttle setting.

In the flying trials to be given in art. 84 an identical range was obtained at 5,000 and 15,000 ft. when flying on a 'weak' mixture, and this agreement is within the experimental errors incidental to such trials in the air.

ART. 83. *Fuel economy in the supercharged engine.*

If the power necessary for driving it were derived entirely from the waste energy of the exhaust gases, the addition of a supercharger would increase the power output of an engine with, at the same time, some slight improvement in its thermal efficiency and fuel consumption per h.p. This follows from the fact that although there would be no increase of the expansion ratio of the engine, upon which the efficiency depends, a part of the extra energy derived from the waste gases would appear as positive work on the crankshaft during the induction strokes; and furthermore the proportion of the heat lost to the cylinder walls would be less at the increased air density.

When the supercharger is driven by gearing from the crankshaft the conditions are much less favourable to efficiency, and such an engine must always compare unfavourably as regards fuel consumption per h.p. with a larger engine giving the same power, but normally aspirated.

An engine driving its own supercharger is equivalent to one in which the total compression ratio has been increased without there being at the same time any increase of the expansion ratio, and even if an expansion ratio equal to the full compression could be provided in the cylinders, such an engine would be less efficient than one in which the whole compression-expansion process took place in a cylinder. Compression by a piston is a very efficient process, considered in terms of the power required to compress and deliver each lb. of air, and the rise of temperature produced approximates closely to that for the adiabatic compression. When the same degree of compression is produced in a centrifugal supercharger the power required is greater in the ratio of about $1/0{\cdot}65$ (see Chapter X), and the extra work done per lb. of air appears as an increased temperature-rise in the air delivered.

The foregoing considerations apply to a supercharged engine

operating at full throttle. If it were not for detonation and the difficulty of getting rid of the waste heat, every supercharged engine might be operated at full throttle at all heights from the ground upwards; and when considering the fuel consumption of an engine in which the supercharger is controlled so as to give a constant manifold pressure up to the rated altitude, it makes for clarity to regard this simply as an engine of a constant total compression ratio (in supercharger+cylinders) which has always to be throttled so long as it is below its rated altitude. Regarded in this way it is clear why the conditions in such an engine should be especially unfavourable to fuel economy, for in addition to the loss of efficiency inherent in the centrifugal compressor there is, below the rated altitude, the increase of the pumping loss associated with a throttled engine.

On the supercharged water-cooled engine referred to in table 54 motoring tests were made at four heights, up to 23,000 ft., at each of the speeds and throttle settings at which the power and fuel consumption had been measured. Ample data for that engine exist, therefore, from which to deduce the effect of altitude upon the rate of fuel consumption both per I.H.P. and per B.H.P.

The figures of air consumption per I.H.P. have already been given in table 54. There was evidence that, at the same average fuel-air ratio, imperfect mixture conditions had been responsible for an increase of about 4 per cent. in the air per I.H.P. at 15,000 ft. as compared with ground-level; but the figures quoted below for the fuel consumption per B.H.P. hour indicate that, at heights above the rated altitude, any increase in the air per I.H.P. is more than compensated by a lower value of the losses, when compared with conditions near the ground.

As was done for the normally aspirated engine, the comparison will be made under conditions corresponding to a constant A.S.I. reading in flight, and therefore a constant B.M.E.P. The latter has been chosen as 65·5 lb. sq. in. at which value one set of observations was actually made in the bench tests under 15,000 ft. conditions. The essential figures are given in table 56. Lines 1, 2, and 4 give actual observations made in the bench tests, while lines 3 and 5 show the calculations for the correct engine r.p.m. in level flight to correspond with a B.M.E.P. of 65·5 lb. sq. in. at each height.

In col. 5 are given the figures for the L.M.E.P., either as they were observed in motoring tests, or corrected for small changes of speed. The effect of the gear-driven supercharger is to turn the engine into one with a total compression ratio which depends on the square of the speed, and the linear relation between the r.p.m. and the L.M.E.P., found for the normally aspirated engine, no longer

TABLE 56

Fuel consumption per B.H.P. hour for supercharged water-cooled engine of rated altitude 9,600 ft. Comparison at three heights at constant B.M.E.P. and corresponding r.p.m. for level flight.

Engine C.R. = 6 : 1

Line	*Height ft.*	*B.M.E.P.*	*r.p.m.*	*L.M.E.P.*		*Min. fuel consumption lb./B.H.P./hr.*		P_T *lb. sq. in.*	P_m *lb. sq. in.*
				Obs.	*Calc.*	*Obs.*	*Calc.*		
1	15,000	65·5	1,804	14·2	14·5	0·445	..	6·9	8·65
2	9,600	69·1	1,576	14·8	..	0·452	..	8·6	9·7
3	9,600	65·5	1,655	..	15·5	..	0·46	7·6	9·2
4	Ground-level	63·2	1,576	20·2	..	0·49	..	8·45	10·00
5	"	65·5	1,430	..	17·5	...	0·47	9·0	10·35

holds. Nevertheless, over the range of conditions which are important, constants can be found for equation (71) which will make it give the L.M.E.P. correct to within about 1 lb. sq. in. The figures obtained from the equation

$$(\text{L.M.E.P.})_{\text{atm}} = \frac{27{\cdot}8}{\sqrt{T}}\frac{N}{N_0}P - \frac{8{\cdot}0}{\sqrt{T}}\frac{N}{N_0}P_T + 0{\cdot}50\frac{N}{N_0} \qquad (84)$$

are shown in col. 6 of table 56 by the side of the observed figures. In applying the equation it should be noted that the pressure P_T is not the manifold pressure P_m, as in the normally aspirated engine, but is the pressure immediately after the throttle and before the supercharger. Both P_T and P_m are given in table 56 in order to illustrate how the speed affects the pressure ratio through the supercharger.

The fuel consumptions per B.H.P. hour under conditions of constant A.S.I. reading are shown in lines 1, 3, and 5 of table 56, as observed at 15,000 ft., and from observations adjusted in accordance with the change of mechanical efficiency of the engine with speed, at the other two heights. These figures are the minimum consumptions obtainable. If the fuel per B.H.P. hour at ground-level and 15,000 ft. is estimated for the chemically correct mixture on the basis of the air consumption, as was done for the normally aspirated engine in table 55, we obtain the results in table 57, assuming a mean figure for the I.H.P./lb. air/min. of 10·4. Table 57 shows a saving of 3 per cent. at 15,000 ft. as compared with that of 5½ per cent., based upon the observed minimum consumptions and given in table 56. The minimum figures were obtained at mixture

strengths 11 per cent. and 16·5 per cent. weak at ground-level and 15,000 ft. respectively, so that a larger margin is to be expected than between the figures of table 57.

TABLE 57

Estimated fuel consumption per B.H.P. hour for a supercharged water-cooled engine of compression ratio 6 and rated altitude 9,600 ft., based upon an assumed constant value of I.H.P./lb. air/min. = 10·4 lb. and a chemically correct mixture.

	At ground-level	*At 15,000 ft.*
B.M.E.P.	65·5	65·5
r.p.m.	1,430	1,804
L.M.E.P.	17·5	14·5
I.M.E.P.	83	80
I.H.P.	195	237
Air lb./min.	18·75	22·8
Fuel lb./min.	1·34	1·63
Fuel consumption lb./B.H.P./hour	0·52	0·505

One may, perhaps, sum up the results of the observations upon the effect of altitude on fuel consumption by saying that, assuming good distribution, the supercharged engine of rated altitude 10,000 ft. is likely to show a saving in fuel per B.H.P. hour at 15,000 ft., as compared with ground-level, of 3–4 per cent. when tested at the same mixture strength and at a throttle opening sufficient to give the same torque at each height; and that the normally aspirated engine under the same conditions may be expected to show a saving of about 2 per cent.

ART. 84. *Air endurance and range.*

It may, according to circumstances, be important for an aeroplane to cover a given distance in a minimum of time, or to cover the greatest possible distance upon a given weight of fuel, and the general question arises whether advantage may be gained by flying at any particular height, and, if so, what are the conditions which determine that height. To supplement the treatment given in Chapter II, where a constant fuel consumption per B.H.P. hour under all conditions was assumed, there are given in this article the results of fuel measurements in level flight at heights ranging from 1,500 to 15,000 ft.

Any flight which is to be economical, either in time or in fuel consumption, must always depend on three different kinds of efficiency:

(1) the aerodynamic efficiency of the aeroplane;
(2) the thermal efficiency of the engine; and
(3) the human efficiency of the navigator.

The wind normally differs both in speed and direction at different heights, and much will depend therefore upon the human efficiency factor in the choice of the best height, air-speed, and direction in which to fly; but, before any proper choice can be made, the necessary data must be supplied in the form of an answer to certain clearly defined aerodynamic and engine problems, and it is with the latter that this book is concerned.

Economy in time is a straightforward matter, for the distance between two points will be traversed in the least possible time by flying at a low height with fully open throttle, if the engine is unsupercharged, or, if it carries a supercharger capable of maintaining its ground power up to a certain 'rated height', then by flying at that height; in each case with the engine running at its maximum permissible r.p.m. Under these conditions aerodynamic and engine efficiencies go by the board, for at high speeds the drag of the aeroplane will be high, and at full throttle an economical fuel-air mixture cannot be used on account of overheating.

It was shown in Chapter II that at any height there is a certain forward speed at which the drag of an aeroplane of given weight is a minimum, and its aerodynamic efficiency therefore a maximum, and that the A.S.I. reading at which this minimum value of the drag occurs is the same at all heights. Since it was also shown that the drag depends only upon the A.S.I. reading, V_i, it follows that the minimum value of the drag itself, D_m, is the same at all heights. It was also shown that except near the ceiling of the aeroplane the most economical speed is a long way off the maximum speed, so that the two problems, of reaching a distant point as quickly as possible, and with a minimum expenditure of fuel, require very different flying conditions.

When flying at the most economical speed the work done per hour at any height would be

$$VD_m,$$

and since this must be equal to the thrust horse-power (T.H.P.) delivered by the engine,

$$(\text{B.H.P.})\eta = AVD_m,$$

in which η is the airscrew efficiency and A is a constant. If p is the rate of fuel consumption of the engine in lb. per B.H.P. hour, then the fuel consumed per hour is given by

$$p \times (\text{B.H.P.}) = p\,\frac{AVD_m}{\eta}. \tag{85}$$

If D_m is measured in lb. and V in m.p.h. then $A = 0{\cdot}00267$.

The fuel consumption per mile at the most economical speed, given by

$$p\frac{AD_m}{\eta},$$

is simply proportional to p, and the minimum fuel consumption per mile will only vary with height in so far as p does so.

In Chapter II the assumption was made that p was constant under all conditions of height and throttle setting, and on this assumption it would follow that the maximum range of an aeroplane in still air would be the same at all heights. The results given earlier in the present chapter have shown that under conditions of a constant A.S.I. reading there is likely to be a small diminution in p of 2–4 per cent. between ground-level and 15,000 ft. provided the mixture is constant, and is an economical one, but that the corresponding gain of 2–4 per cent. in the maximum range obtainable may easily be nullified if, owing to unfavourable carburation conditions, the average fuel-air ratio has to be richer under the colder conditions.

In the present article the results of fuel consumption measurements in flight[66] will be discussed in which the range in miles per gallon was measured at four different heights for an aeroplane with a normally aspirated engine. From the observed effect of height upon the range one can deduce the relative, but not the absolute, values of p at different altitudes.

Three series of trials were made, the first with no special precautions to secure an economical fuel-air mixture; the second with the mixture-control used so that the average mixture supplied to the engine was the weakest on which it would maintain full power for each particular throttle setting and speed, i.e. would maintain its proper r.p.m. when flying level; and the third series with the weakest mixture on which the engine would run satisfactorily. This last mixture was sufficiently weak to cause a drop of engine r.p.m. in level flight of 3 per cent. below normal. Although satisfactory as a practical criterion of what is possible in the way of weak mixtures in flight, this use of engine behaviour as the indicator gives no definite information of what the air-fuel ratio was at the different heights, nor whether it was the same. As indicated in art. 83 there is reason to expect that with the lower air temperatures at high altitudes unsteady running and loss of power would set in at a lower average air-fuel ratio than lower down.

The results of the first series of trials were rather erratic, as would be expected, and are not reproduced in detail. The maximum range at all heights above 1,500 ft. was about 7 m.p.g., but the optimum value for V_i varied from 70 to 82 m.p.h. The results of the other

two series are illustrated in figs. 143 and 144. It will be seen, in the first place, that the value of V_i which gave the maximum range was, for each series, exactly the same at all heights, as predicted by theory. It was 72 m.p.h. in fig. 143 and 75 m.p.h. in the trials with the weakest possible mixture. As compared with 7·1 m.p.g. in the first series, with no special precautions taken, the maxima in the

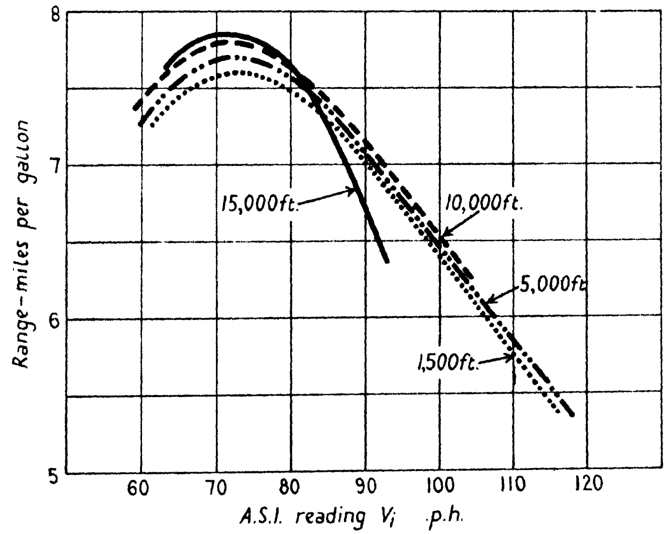

FIG. 143. The effect of height and speed on observed range with a normally aspirated engine. Mixture control used so as to give the weakest fuel-air ratio consistent with no loss of power. Compression ratio 6·3, engine always throttled below 5,000 ft.

second series range from 7·6 to 7·84, and in the third from 8·0 to 8·15 m.p.g., according to the height.

In the second series (fig. 143), that is, the one with the weakest mixture for maximum power, there was a small but steady increase of the maximum range with height amounting to 3 per cent. between 1,500 and 15,000 ft., which closely confirms the estimates of the last article. In the weakest mixture series (fig. 144), there was an increase of 2 per cent. for the same change of height, but the trials at 5,000 ft. gave a range equal to those at 15,000. The mixture in the third series was weak enough to make the running of the engine unpleasantly rough, and sometimes unsteady, and the bench tests have shown that when this is so the performance of an engine is much affected by the condition of the sparking plugs. In these circumstances 2 per cent. is probably the limit of accuracy to be expected in the flying trials.

These trials, moreover, were all done with a constant ignition advance of 42°, and one of the conditions for obtaining consistent results on very weak mixtures was thereby not fulfilled, for it is essential that the ignition advance should be that giving maximum power. In some later trials[70] on a very similar engine, but in a different aeroplane, the effect of ignition timing has been investigated and the importance of an adequate advance, for economical working in

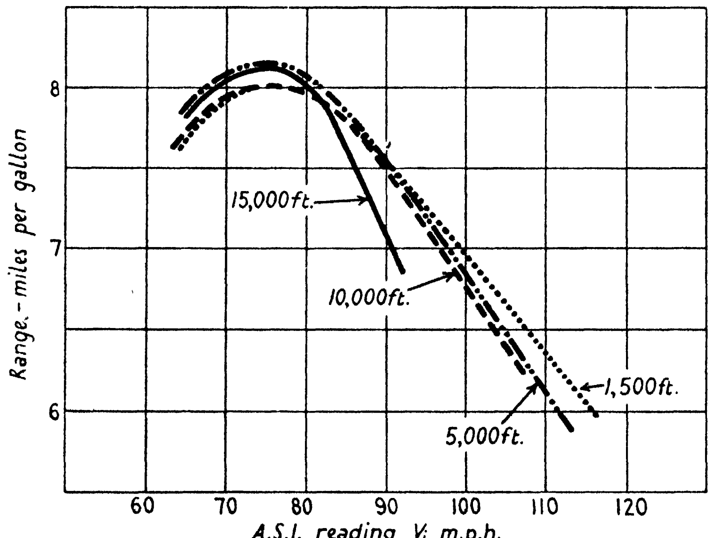

FIG. 144. Effect of height and speed on observed range when the mixture control was used so as to produce a drop of 35–55 r.p.m. due to weakening of the mixture. Owing to unsteady running it was not always possible to work with the full drop of r.p.m. Same engine as fig. 143.

a throttled engine, has been fully demonstrated. The results, expressed in miles per gallon at 2,000 and 12,000 ft., are shown in fig. 145. In all these trials the mixture was nominally the same as in those of fig. 144, i.e. it was that which caused a drop of 3 per cent. in engine r.p.m. when flying level. The values of the maximum range, however, must not be compared with those of fig. 144, because of a difference in the gross weight of the aeroplane.

Compared amongst themselves, the curves of fig. 145 show a substantial increase of the maximum range with increase of ignition advance. It amounts at all heights to 7 per cent. for the full change from 35 to 51 degrees. In this connexion it may be mentioned that the maximum range was obtained at values of the engine r.p.m. equal to about 1,500 at 2,000 ft. rising to 1,700 at 12,000 ft. It is of

interest to note that the effect upon the range of retarding the ignition from 51 deg. to 35 deg. before the dead-centre was more detrimental at 2,000 ft. than at 12,000 ft. This is in conformity with the test engine results given in fig. 40 (i), where the necessary ignition advance for a high thermal efficiency was shown to increase with the amount of throttling. Maximum range at 2,000 ft. would, of course, be obtained with the engine much more throttled down than it would be at 12,000 ft.

On the question of the influence of height upon range, these trials

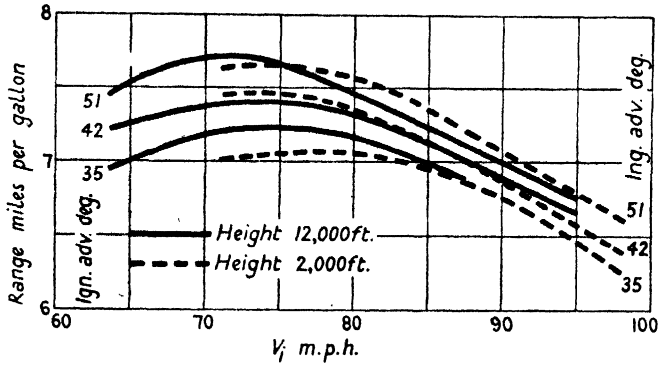

FIG. 145. The effect of ignition timing on range. Fuel-air mixture weakened to give 3 per cent. drop of engine r.p.m., flying level.

suffer from the length of time over which they extended and the variability of atmospheric conditions. Besides the two series in fig. 145, which were done at about the same time and give nearly the same figure for the maximum range at 2,000 ft. and 12,000 ft. there was also a series at 5,000 ft., done about a year earlier, which gave a maximum range 3 per cent. higher than the best achieved afterwards. In the circumstances this is probably the limit of accuracy of the tests, and one can only conclude that they gave no evidence of any variation of range with height. In the series at 5,000 ft. there was evidence of the favourable effect on range of a high air temperature, resulting in better carburation; and one is led to the conclusion that the calculated specific fuel consumption—lower by 2 per cent. at 15,000 ft. than at ground-level—would represent the real facts if a uniformly weak fuel-air mixture could be maintained throughout the engine, but that under practical conditions the corresponding advantage in maximum range is not achieved by flying high because of less favourable conditions of carburation, which lead to unequal fuel distribution between cylinders and nullify the effect of the reduced L.H.P. at altitude.

The rapid fall off in range in figs. 143 to 145 as the A.S.I. reading was increased beyond about 80 m.p.h. reflects the fall of aerodynamic efficiency as the drag increased; for the engine efficiency would have been almost unaffected, a reduction of the pumping loss by opening of the throttle being offset by an increase, with speed, of both the pumping and the mechanical friction loss. The exceptionally rapid fall of the curve for 15,000 ft. in the earlier trials, however, must reflect some abnormally rapid increase in the rate of fuel consumption per B.H.P. hour in the engine, which can only be attributed, in the absence of fuller information, to unsatisfactory carburation under high altitude conditions. There is no sign of the same effect in the curve for 12,000 ft. in fig. 145.

APPENDIX I

A note on dimensions.

CONSIDER two engines of geometrically similar form but of different sizes, with their linear dimensions in the ratio $L : 1$. The rotational speeds of the two will be limited by the stresses set up in the bearings, due to inertia forces. Let the ratio of the maximum permissible r.p.m. be as $1 : n$, then

$$\text{inertia forces } (Mr\omega^2) \propto L^4/n^2,$$

and

$$\text{bearing stresses } \propto L^2/n^2.$$

The bearing stresses will, therefore, be the same so long as L/n is constant, i.e. so long as the mean piston speeds are the same.

Since piston and valve areas are each proportional to L^2, the volumetric efficiencies will be the same at equal piston speeds and bearing stresses.

If the two cylinders are also of the same thermal efficiency, then their powers at the same bearing stresses and piston speeds will be proportional to the piston areas, and therefore

$$\text{power output} \propto L^2,$$

and

$$\text{h.p. per litre} \propto 1/L.$$

It follows, also, that in so far as the weight of an engine can be taken as proportional to L^3, the weight per h.p. is proportional to L. It should diminish, therefore, with the linear dimensions among engines of equal merit. It is well to keep in mind this fundamental relationship when comparing engines of widely different size, but as applied to aero-engines it cannot be pressed very far, for the reason mentioned in art. 1, that so much of the necessary weight of a light engine is not subject to dimensional treatment.

APPENDIX II

The assumptions involved in the calculations of the figures of table 3.

FROM observations of the volumetric efficiency of the normally aspirated engine one may conclude that the amount of heat received by the ingoing charge from cylinder walls, piston, and valves is sufficient to raise its temperature about 45° C., say from 15° to 60° C. At a compression ratio of 5 : 1 this temperature is raised by admixture of residual gas to 122° C. (table 20 (i)).

In passing through a supercharger the mixture temperature will

be raised by more than that corresponding to adiabatic compression. The 'adiabatic temperature efficiency' of the centrifugal supercharger may be taken as 0·65 and the rise of temperature through it, accordingly, as 1·54 times the adiabatic rise. The temperature of the air as it leaves the supercharger is thus calculated from the pressure ratio and is given in col. 3 of the table below. It is assumed for simplicity that all evaporation of petrol occurs after leaving the supercharger.

TABLE (Appendix II)

Compression ratio	*Pressure ratio through supercharger*	*Air temp. leaving supercharger*	*Charge temp. before heating by residual gas*	*Ratio by weight* $\frac{\textit{residual gas}}{\textit{fresh charge}}$	*Mixture temp. before compression*
5 : 1	1*	15° C.	60° C.	0·074	122
	1·25	42	77	0·060	127
	1·50	69	94	0·051	136
	1·75	92	107	0·045	144
	2·0	110	115	0·039	147
6 : 1	1*	15	60	0·060	111
	1·25	42	77	0·049	118
	1·5	69	94	0·041	128
	1·75	92	107	0·036	136
	2·0	110	115	0·032	141

* Normally aspirated condition, see table 20 (i).

Now the rise of temperature of the charge from 15° to 60° in the normally aspirated engine is due to heat picked up from cylinder walls, piston, and valves. At a supercharge pressure of 2 atm., when the air enters the induction system at 110° C., there will clearly be much less gain of heat, and we have to make some working assumption as to the amount.

As a rough approximation it has been assumed that the rise of temperature after leaving the supercharger would be 10° C. less for each 0·25 atm. of supercharge, so that at 2 atm. delivery pressure the mixture will be at 115° C. before mixing with the residual gas, and for lesser degrees of supercharge the figures are as in col. 4 of the table.

In calculating the temperatures after mixing with residual gas, allowance has to be made for the increasing weight of fresh charge and for the fact that the mean volumetric heat of the residual gas while cooling from the assumed temperature of 850° C. will be greater than that of the fresh charge. With these allowances made the figures in the last two columns of the table are obtained.

APPENDIX III

Derivation of connexion between ψ and ϕ, the power factors for I.H.P. and B.H.P.

$B =$ B.H.P., $L =$ L.H.P., and $I =$ I.H.P. Suffixes 0 and h refer to ground-level and altitude h.

$$B_h = I_0 p\theta^{-\frac{1}{2}} - L_h$$

$$L_h = \lambda L_0 + (1-\lambda) L_0 p\theta^{-\frac{1}{2}}$$

$$L_0 = \frac{1-m}{m} B_0.$$

$$\therefore\; L_h = \frac{1-m}{m}\lambda B_0 + \frac{(1-\lambda)(1-m)}{m} B_0 p\theta^{-\frac{1}{2}},$$

whence $$B_h = p\theta^{-\frac{1}{2}}\frac{B_0}{m} - (1-m)\lambda\frac{B_0}{m} - (1-\lambda)(1-m)\frac{B_0}{m}p\theta^{-\frac{1}{2}},$$

whence $$\phi = \frac{B_h}{B_0} = \left[p\theta^{-\frac{1}{2}}\left(1+\frac{\lambda-\lambda m}{m}\right) - \left(\frac{\lambda-\lambda m}{m}\right)\right],$$

or $$\phi = \psi\left(1+\frac{\lambda-\lambda m}{m}\right) - \frac{\lambda-\lambda m}{m}.$$

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